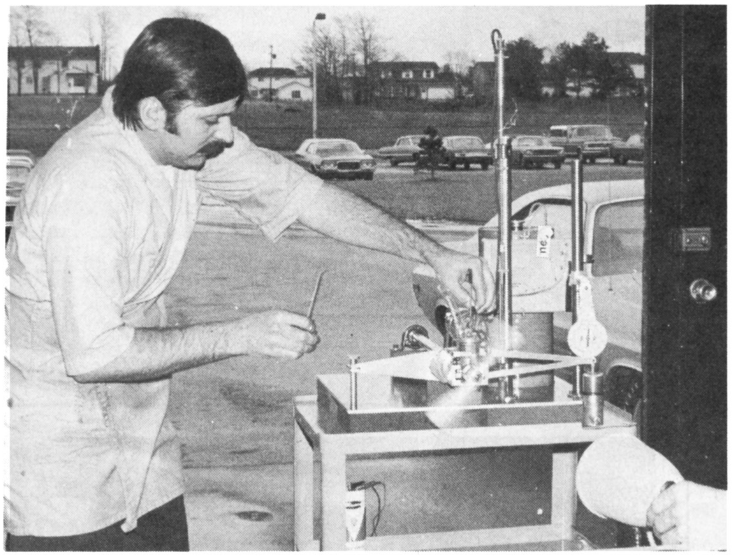
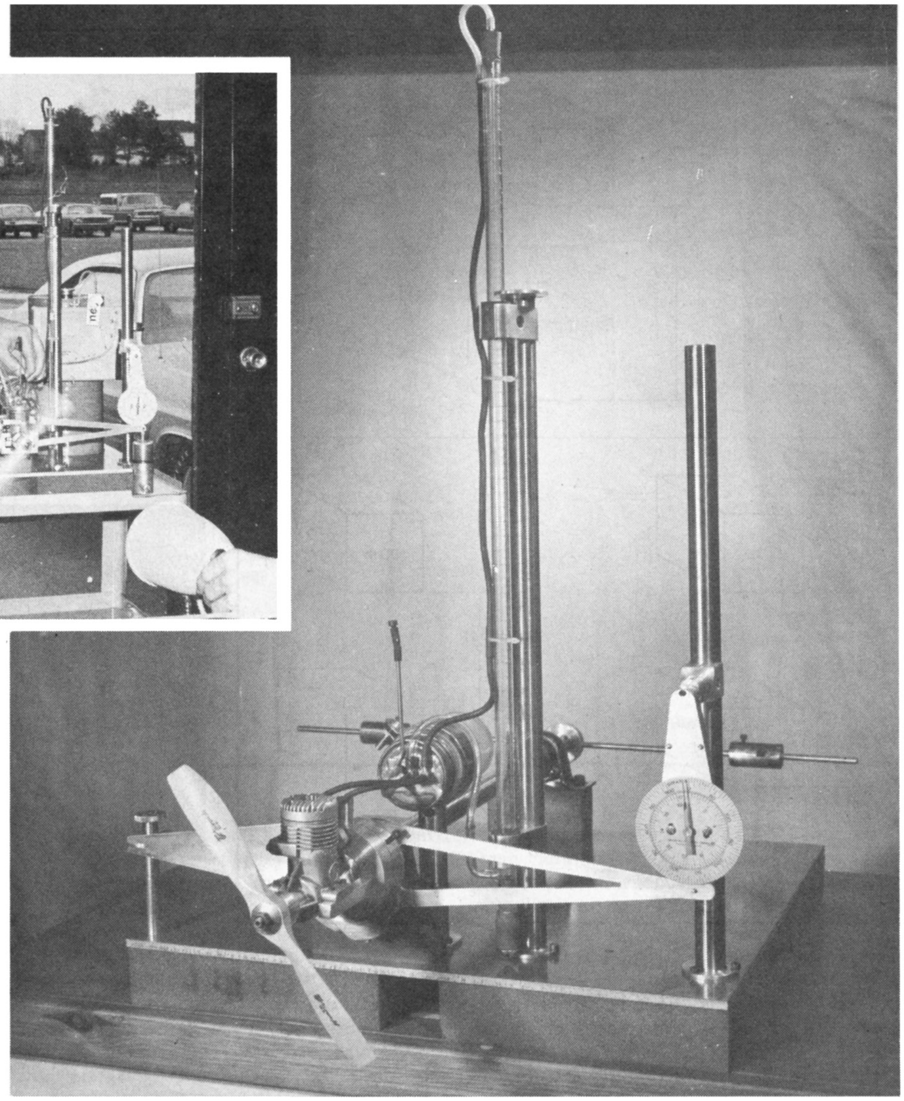


Dave's about ready to switch from tank feed to the Buret for the fuel consumption measurement. At right: Basic Dynamometer Instrument. Notice force measuring scale; a fuel measuring Buret, shock absorbing base. Engine mounts on machine.



Photos by the Author

## Part 1

# DYNAMOMETER

## AN ENGINE PERFORMANCE ANALYSIS

**Part I of a new series which explores the potentials of an engine as never before. Myths explode as science takes over.**

**by Dave Gierke**

In this initial installment we will discuss the following concepts and principles necessary for understanding engine performance analysis:

A – Work
B – Power
C – Torque
D – Horsepower
E – Energy
F – Efficiency

When these terms are used in a scientific sense, their meanings are sometimes quite different than is normally accepted in everyday conversations. The scientific meanings of these terms as used by a technician or engineer will be discussed here, giving us a basis from which we may develop an understanding of the analysis procedure. As you shall soon see, our purpose is to *identify* these terms and learn how they may be measured.

The production of power for the purpose of doing work of one kind or another, is the principal reason for any engine's existence. In order to compare the output of one engine to another or to determine how well it is functioning, it is necessary to have some method of measuring the power and expressing the results in uniform terms or parameters. A parameter is an expression in which a number of variables are combined into a factor which can be used as an indicator of performance.

The "power development sequence" as described below is designed to acquaint you with these essential engine performance factors which we will put to practical use later on:

A. **WORK.** We are all familiar with the everyday meaning of this word, however, the meaning is quite different when used as a scientific term. Trying to push a heavy piano from one side of the room to the other that won't move is called "difficult work" in everyday language. You push hard, become exhausted, but the piano does not move. According to the scientific definition, no work has been performed. Work is accomplished only when a force moves through a distance. The amount of work done is described as follows:

**WORK** is the amount of force multiplied times the distance moved in the direction the force acts.

We use the letter "F" to represent force, "W" for work and "d" for distance. The following equation for work may be derived:

Work=Distance x Force *or* W=d x F

If we have a 100 pound weight and raise it five feet off the ground, work has been done, as shown in Figure 1.

How much work has been done and how do we measure it? Using the equation for work, subtitute the given values:

Example – W=d x F
W=5 Ft. x 100 Lbs.
W=500 Ft. Lbs.

Since we multiplied *feet x pounds*, we show the result as "Ft.-Lbs." of work. Work is done when anything mechanical is performed, such as lifting weights, stretching springs, or compressing a gas. Remember that a *force* must act through a *distance*. This concept will come into play shortly, when we discuss the measurement of engine power.

B. **POWER.** *Time* is not involved with the concept of work. It makes no difference whether one second or ten seconds were taken to lift the before-mentioned weight. The identical amount of work was accomplished. Now we need a term which will describe how fast work is done. In most practical applications as well as scientific matters, the time required to perform an amount of work is important. Example: When a clock spring is wound, it requires only a few seconds, but the mechanical energy may require days in which to be

Fig. 1

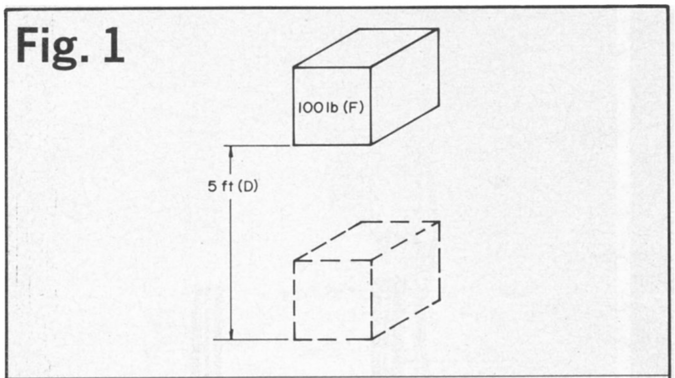



Fig. 2

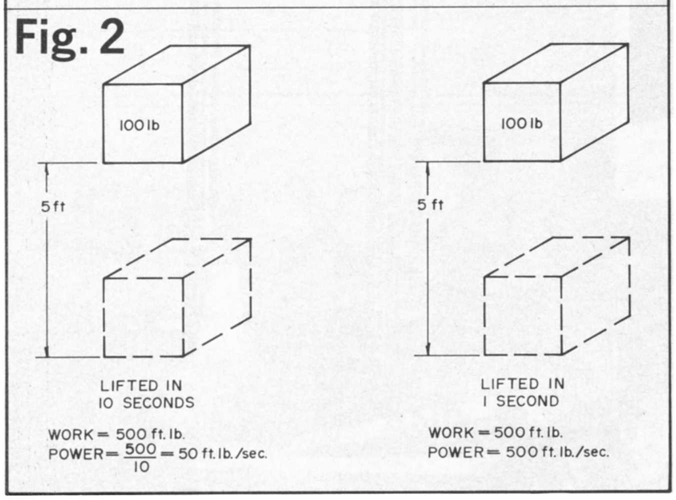
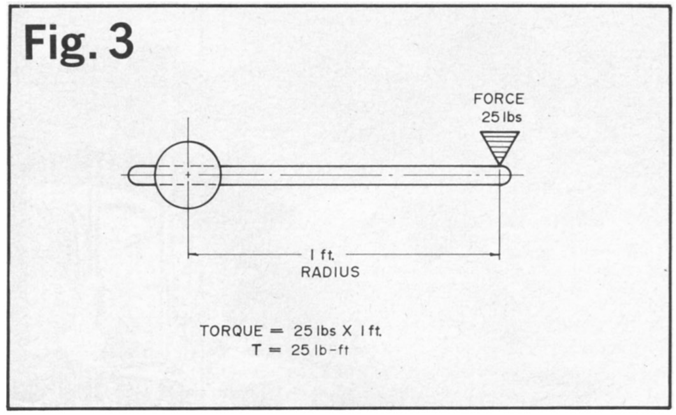
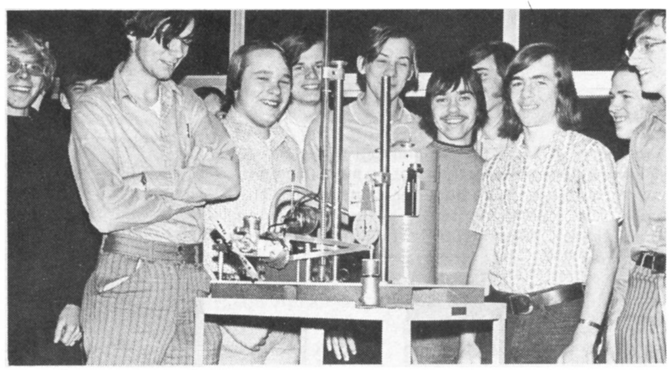



Fig. 3



The Power Technology students gather around the Dyno. Who's going to use the "chicken stick" today? (Idea makes a non-modeler fearful-fingered.)

expended. *Power* is the term used to describe the amount of *time* necessary to do *work*.

*Power* is the *rate* of doing work, or the rate at which energy is utilized.

To figure *power* (P)—divide *work* (W) by *time* (T). This can be shown as a basic equation:

$$\text{Power} = \frac{\text{Work}}{\text{Time}} \quad \text{or } P = \frac{W}{T}$$

Power is described as foot-pounds (work) per second (time).

$$\text{Power} = \frac{\text{Feet x Pounds}}{\text{Seconds}}$$

Using the above equation for Power—A 100 Lb. weight was lifted five feet in 10 seconds, the power output is:

$$\text{Power} = \frac{\text{5 Feet x 100 Lbs.}}{\text{10 seconds}} = \frac{\text{500 Ft.-Lbs.}}{\text{10 Sec.}}$$

$$= 50 \ \frac{\text{Ft.Lbs.}}{\text{Sec.}} \ \text{Answer}$$

But, if the weight had been raised in one (1) second, the power output would be as follows:

$$\text{Power} = \frac{\text{5 Feet x 100 Lbs.}}{\text{1 Second}} = \frac{\text{500 Ft.-Lbs.}}{\text{1 Sec.}}$$

$$= 500 \ \frac{\text{Ft.Lbs.}}{\text{Sec.}} \ \text{Answer}$$

The work in both cases is the same. The power output, as you can see is ten times greater in the latter example. See Figure 2.

C. **TORQUE.** Until now, we have discussed *power* in terms of *linear motion*. Now, however, as is the case with most of our engines, the output is rotational and must be measured as such. It is possible to measure rotational power, however, the concepts differ somewhat.

When rotational power produces a twisting force, we call this *torque*. We do, however, measure this torque or twisting force in a similar manner to that of *work*.

*Torque* equals *force* (F) times radius (r). The term *force* is used the same as it is in measuring work. The *radius* (r) is the distance from where the force acts to the center of rotation of the shaft.

$$T = F \text{ x } r$$

The units of measurement for *Torque* can easily be confused with the units of measurement for *work*, because they are quite similar.

*Torque* is always identified as *force x distance*—(Lbs.-Ft.; Lbs.-In.; Oz.-In.). See photos.

*Work* is always identified as *distance x force*. There are many confusions surrounding these two identifications, one of which seems to be incorrect identification on most brands of torque wrenches:

Ft.-Lb. or In.-Lb.

Note the following example showing how to calculate torque being placed on a bolt head while it is being tightened. See Figure 3.

If a nut cannot be turned by the wrench, there has been no *work* accomplished because the *force* has not traveled through a *distance*. However, another nut which can be turned by the wrench with a *constant force* of 25 Lbs. on the handle, for one complete turn is performing *work*. The distance traveled, in this case, is the circumference (C) of the circle, of rotation, of the force.

*Example:* A *force* of 25 Lbs. was exerted at the end of a one foot long wrench. The wrench was turned one complete rotation. What was the work done?

Distance=Circumference=$2\pi r$=
2 x 3.14 x 1=6.28 Ft.
W=d x F=6.28 Ft. x 25 Lbs.=
158 Ft.Lbs. (W)

D. **HORSEPOWER.** James Watt defined *one horsepower* as the ability to do *33,000 Ft. Lbs. of work per minute or 550 Ft. Lbs. of work per second* Watt had found it necessary to compare his improved steam engine with the number of horses it could replace. He undertook this by experiments to determine the power of an average horse. To be on the safe side and not overrate his engines, he made his unit of power more than what an average horse is able to exert.

So, we can easily see that one horsepower is a 550 times larger unit of power than a foot-pound per second. Horsepower has become our standard unit of power measurement since the 18th century experiments of Watt.

In order to measure Horsepower, we generally incorporate an instrument known as the *Dynamometer*. As we shall see later, there are many types of dynamometers but all of them are used to *indirectly* determine horsepower by measuring *Torque* and the *R.P.M.* of the engine.

Recall that we have already determined that *power* can be measured both if it is *linear* or *rotary*. As before, if a torque wrench can be turned through one complete revolution at a constant force, *work* can be measured.

W=d x F
W=Circumference x F
W=$2\pi$ r x F

From the above we may go a step further and determine *power* which is the *rate of doing work*, or:

P=Work x Revolutions per Minute

$P = W \times R.P.M.$

$P = 2\pi \; R \times F \times R.P.M.$

This can now be reorganized into units of *Torque*, R.P.M. and a constant ($2\pi$):

$P = F \times r\,(\text{Torque}) \times R.P.M. \times 2\pi$

$P = \text{Torque} \times R.P.M. \times 2\pi$

Finally, to find *horsepower* divide the above equation by the number of power units (33,000 Ft.Lb./Min.) in one horsepower. This power unit was selected because the equation is written in terms of *R.P.M.* (revolutions per minute). We must use the horsepower unit for *minutes* rather than *seconds*. They are the same:

60 Seconds x 550 Ft.Lb./Sec.=33,000 Ft.Lb./Min.

The equation for *horsepower* is therefore shown below:

$$H.P. = \frac{\text{Power}}{33{,}000 \text{ Ft. Lb./Min.}}$$

As stated before, *horsepower* is a derived function. It cannot be measured directly. The two factors we can measure are the *Torque* and *R.P.M's* produced at the crankshaft.

If an engine's crankshaft were stationary when it operated, its torque could easily be measured by a torque wrench. However, the shaft is rotating at high speed and it is more difficult to measure the torque.

Almost all dynamometers measure the torque of an engine by converting the rotating torque into stationary torque. This stationary torque is measured by a force measuring scale (as with my machine), a hanging weight (balance beam type) or any one of many other force-measuring devices at the end of the torque arm.

Some of the dynamometer devices used for changing a *rotating torque* to a *stationary torque* are shown in Figure 4.

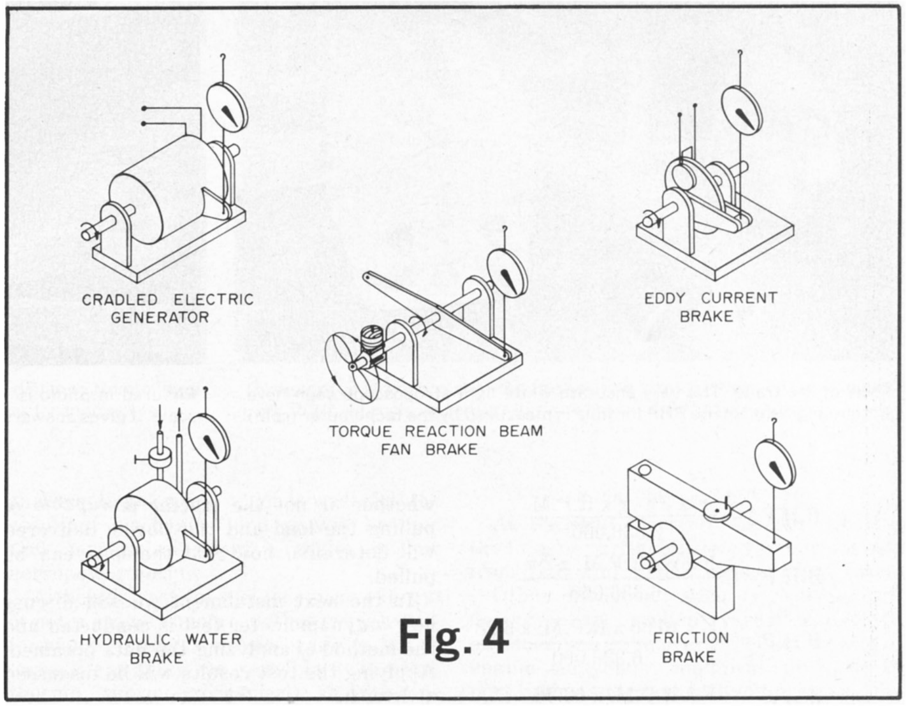



**Fig. 4**

## Power Absorption Unit

The engine analysis system dynamometer which I have developed with the invaluable help of my good friend, Al Mahaffey, uses the "fan brake" system to change rotating torque to stationary torque and absorb the mechanical energy. The fan brake unit (see photo) consists of a series of graduated propellers used to "load" the engine down on the R.P.M. range. I use seven (7) propellers ranging from an 11-7 stunt prop to a cut-down 9-6 "Super M" which has a diameter of 7 inches. These "load" to "unload" props have been tailored to a typical high performance .40 engine. Smaller displacement engines would require smaller "loads." Obviously, the reverse would be true concerning larger engines (.60).

The engine to be tested is mounted securely on the pivot of a "torque reaction beam" which has a spring scale attached at one end. The distance from the center of rotation to the attachment point of the force measuring system (scale) has been carefully calculated. This radius (r) is "set" so that the formula (B.H.P.) is simple to work with. i.e. B.H.P. (Brake Horsepower)=

$$\frac{F \text{ (in ounces)} \times R.P.M.}{100{,}000}$$

for our machine.

The Torque being measured at any given R.P.M. is controlled by the particular "load" being used. It should be noted that the Torque produced by the rotating engine crankshaft is displayed on the stationary torque reaction beam and scale in the opposite direction from that of rotation. This is adequately explained by Newton's third law of motion: "For every action, there is an equal and opposite reaction."

The manner in which the Brake Horsepower curve of an engine on test is determined may now be somewhat evident. The tests are concerned with fitting the engine with different loads, in turn and measuring the corresponding R.P.M. and Torque figures. The product of these related factors then display themselves as a smooth (more or less) curve. This curve, as you may have guessed, is known as the *Brake Horsepower* Curve which is plotted against R.P.M. See example—Figure 5.

## Torque Measuring System

Our engine analysis system dynamometer is designed to have an effective torque arm radius of 10.08 inches. The denominator in the horsepower equation becomes 100,000 for, as mentioned previously, simpler calculation.

The development of the radius arm (r) and resultant simplified B.H.P. formula is shown below:

B.H.P.=33,000 Ft. Lb./Min.

B.H.P.=6,336,000 In.-Oz./Min.

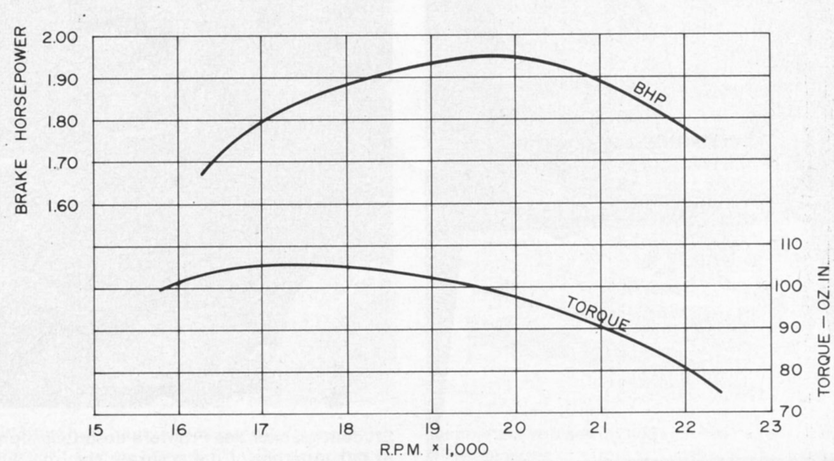
$$B.H.P. = \frac{(1)\ F \times d \times T}{33{,}000} \quad \text{or} \quad \frac{(2)\ F \times d \times T}{6{,}336{,}000}$$

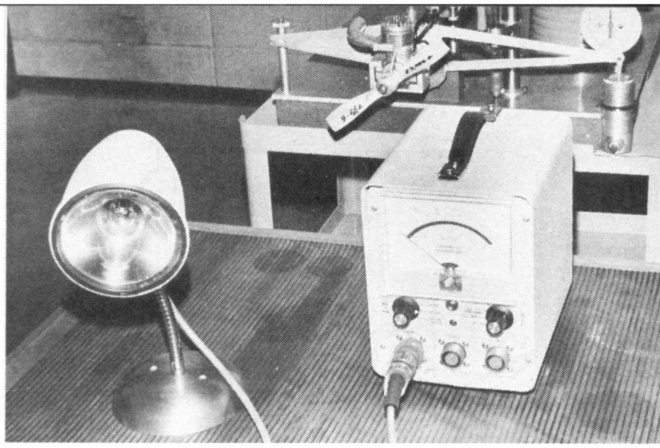
(1) F=Lb. d=Ft. T=Min.

(2) F=Oz. d=In. T=Min.

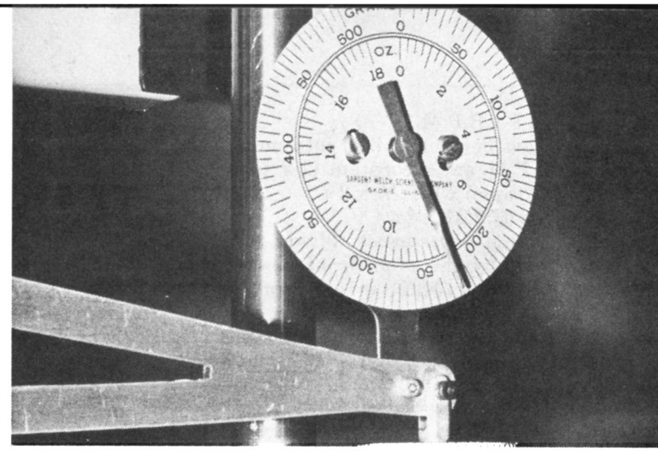
**Fig. 5**

| LOAD PROP | R.P.M. | TORQUE (OZ. IN.) | BHP |
|---|---|---|---|
| 10-6 | 16,300 | 103.0 | 1.67 |
| 91/2-6 | 17,000 | 105.0 | 1.79 |
| 9-6 | 19,500 | 100.0 | 1.95 |
| 81/2-6 | 20,000 | 97.0 | 1.94 |
| 7-6 | 21,800 | 83.0 | 1.80 |





Tools of the trade. The very accurate white light stroboscope seen here. A "time segment" of the BHP formula is measured by the tachometer (rpm).



Pictured in photo is the force measuring device in the form of a spring scale. It gives answers you can understand and compare to known values.

$$\text{B.H.P.} = \frac{\text{F x } 2\pi \ \text{ r x R.P.M.}}{6{,}336{,}000}$$

$$\text{B.H.P.} = \frac{\text{F x r x R.P.M. x } 2\pi}{6{,}336{,}000}$$

$$\text{B.H.P.} = \frac{\text{F x 10.08 x R.P.M. x 6.28}}{6{,}336{,}000}$$

$$\text{B.H.P.} = \frac{\text{F x R.P.M. x 63.36}}{6{,}336{,}000}$$

$$100{,}000 \text{ B.P.H.} = \text{F x R.P.M.}$$

$$\text{B.H.P.} = \frac{\text{F x R.P.M.}}{100{,}000}$$

One of the things that confuses many people when looking at torque-horsepower curves is the fact that torque decreases with increasing R.P.M. We believe that the observer confuses torque with horsepower. Although an engine in question may produce greater power at higher speeds (R.P.M.), its ability to turn a substantial load at these speeds is reduced, calling for smaller propellers (less load).

This problem may be cleared up by restating the definition of torque and horsepower in relation to engine testing. **TORQUE** is the *capacity* of an engine to do work while **POWER** is the *rate* at which and engine does work. A simple example of this would be a tractor pulling a heavy load. The torque developed will determine whether or not the tractor is *capable* of pulling the load and the power delivered will determine how *fast* the load can be pulled.

In the next installment we will discuss how a dynamometer test is conducted and the method of analyzing the data obtained. Applying the test results will be discussed at length.

## E. ENERGY

Thus far, we have discussed *work; power; torque,* and *horsepower,* gradually building an understanding of engine analysis through *dynamometer* testing. Before too long we will be discussing *energy* flow through the engine. *Air-Fuel* flow instrumentation offers a means of studying energy transfer, and will be discussed at length. This requires covering some basic information concerning energy.

Questions such as: "Where does power come from?" "What do you call work that has been stored up to be used later?" The answer to both questions is, in part—**ENERGY.** For our purposes, energy may be defined as:

ENERGY, in any form, is the capacity to do work.

Energy and Work are measured in the same terms or units because energy is the result of work that has been accomplished or is the ability to accomplish work.

Energy exists in many forms. Thermo and nuclear are examples of but two. Mechanical energy is measured in the amount of work a body can perform.

A stretched spring must possess energy, because by releasing it to its original position, work can be performed. An example concerning the spring would be to raise a weight or to compress a gas or pull a lever.

While stretched, the spring has *potential energy,* energy of position. Chemicals such as gasoline have considerable energy or capacity to accomplish work. Chemical energy such as our nitro-methane fuels can be released as *thermo energy* when it is burned with oxygen. In our system (English system of measurement) the energy content in fuels is usually stated in *British thermo units* and abbreviated as *B.T.U.* The B.T.U. is simply another measurement unit.

B.T.U.—the amount of energy required to raise the temperature of one pound of water one degree Fahrenheit.

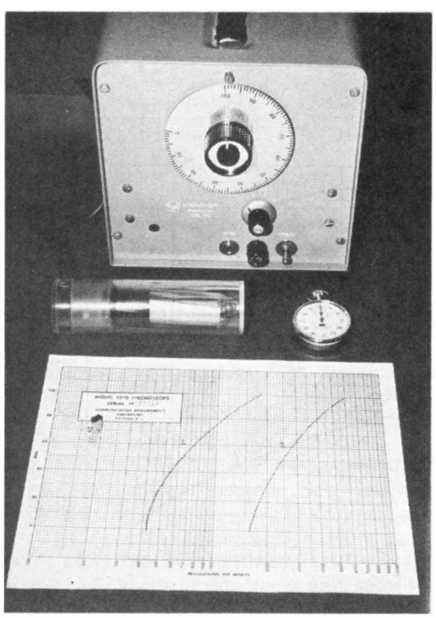
Energy, however, is not always in the form of heat. The mechanical equivalent of one B.T.U. can be shown in Ft.-Lbs.:

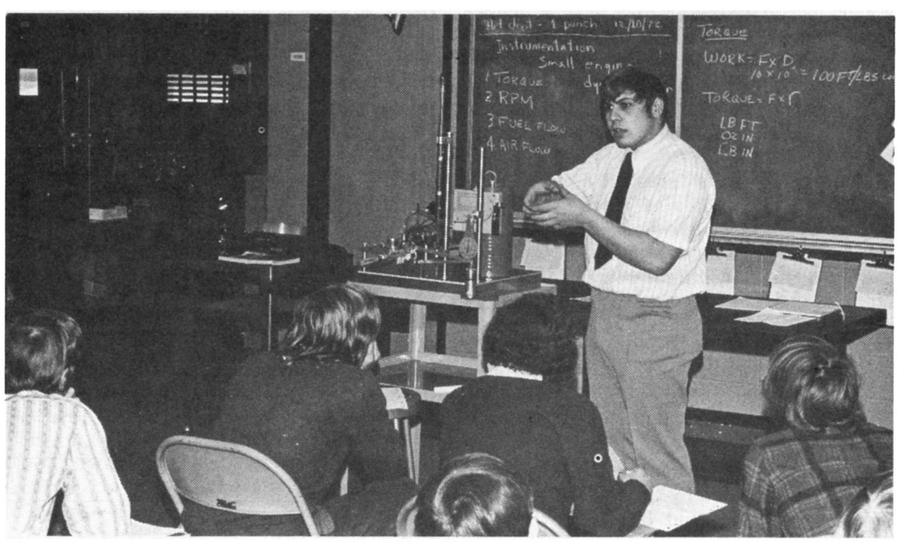
1 B.T.U.=778 Ft.Lbs.

It should be mentioned that power produced by an engine may be expressed in B.T.U./Sec or Ft.-Lb./Sec, or as a horsepower. This conversion is:

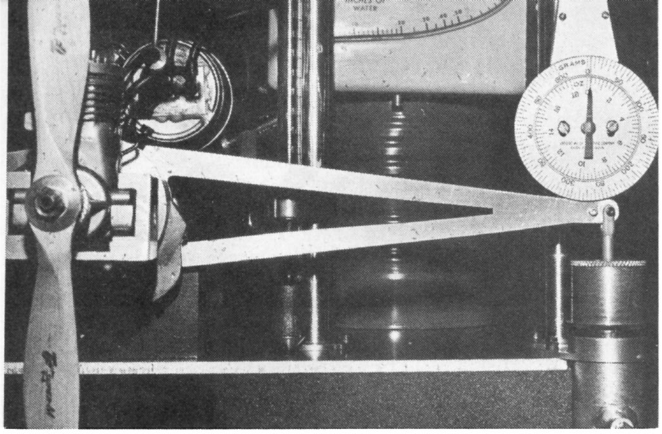
1 H.P.=1.41 B.T.U./Sec

The energy potential of fuels is express-

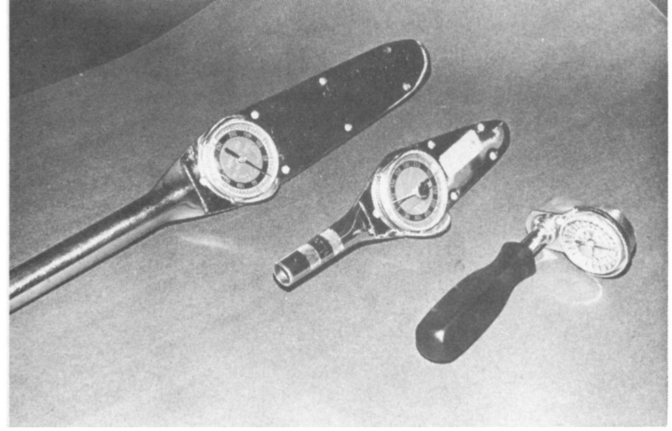




Student teacher Joe Frontera describes a phase of energy systems analysis using the Dynamometer. At left: Another of the accurate rpm measuring instruments is the neon tube strobe pictured here with a stop watch, which is used to measure time required to consume a volume of fuel (10 ml.)



The "torque arm" is clearly shown here as being the distance from engine shaft to the spring scale. (T=f x r). All static and quiet for moment.



Three torque wrenches showing the relative sizes of their measurements. Left: lb./ft. wrench, in center: lb./in. wrench, right: oz./in. wrench.

ed in B.T.U./Lb. Even though most of our fuels are liquids, it is required to measure them in terms of mass because a volume measure such as the quart or gallon changes with temperature. A pound of gasoline can release about 20,000 B.T.U. when transformed (burned) to thermo energy. From the above mechanical equivalent for mechanical energy, the following work potential may be obtained:

$$W = \frac{20{,}000 \text{ B.T.U.}}{\text{Lb. Fuel}} \times \frac{778 \text{ Ft.-Lb.}}{\text{B.T.U.}}$$

$$= 15{,}560{,}000 \text{ Ft.-Lb.}$$

All engines operate on the principle of energy conversion. Our engines which also operate on this principle (chemical-thermo) are referred to as *heat engines*. As a result of this initial energy conversion, a third transformation to mechanical energy performs the work for which the machine was designed:

Chemical—Thermo—Mechanical

It should be mentioned at this time that the energy content of our chemical fuels is directly proportional to the horsepower output of our heat engines.

**F. EFFICIENCY**

In our discussion of engine analysis principles involving work, power and energy, it is necessary to know how efficiently energy is transformed from one type to another. In an engine which is wasteful, the resultant mechanical energy will be low. Low mechanical energy results in low horsepower output.

Although waste of fuel (chemical energy) is not a prime concern with our high performance engines, we do want as much energy extracted as possible (Chemical-thermo; thermo-mechanical), in order to produce more usable power. Here, apparently, is one advantage of the *Schneurle* port engines which are doing so well lately. They make better use of the chemical energy supplied to the cylinder, by more efficient energy extraction.

Efficiency is the ratio of the energy supplied to the work produced. Described another way: For the amount of energy supplied to the engine, how much work did the machine deliver?

$$\text{Efficiency (\%)} = \frac{\text{Work Output}}{\text{Energy Input}} \times 100$$

All machines dissipate some energy in the form of heat and friction losses (thermo and mechanical). In an internal combustion engine, only about 10-25% of the energy supplied is converted into useful work.

An engine working at 25% efficiency produces the following amount of work per pound of gasoline (20,000 B.T.U./Lb.):

$$\frac{15{,}560{,}000 \text{ Ft.-Lbs.}}{1 \text{ Lb. Gasoline}} \times .25$$

$$= 3{,}890{,}000 \text{ Ft.-Lb. Work}$$

As we shall see later, we can calculate the total or partial efficiency of our engines from AIR and FUEL consumption rates.

Other efficiencies such as *Scavenge efficiency* is an excellent yardstick used to measure and compare the effectiveness of engine component modifications such as port timing and crankcase compression, etc.

The general idea again is, however, to increase the total or overall efficiency of the engine as much as possible, in order to utilize and extract as much energy as is practical.

To summarize this material, we might say that we now have some basic understandings of such terms as work, power, torque, horsepower, energy and efficiency. These factors are important if we are to understand the use of the small engine dynamometer and related instrumentation such as the *tachometer, manometer* (air flow rate) and *fuel flow buret.*

In the next installment we will discuss the more practical aspect of engine analysis:

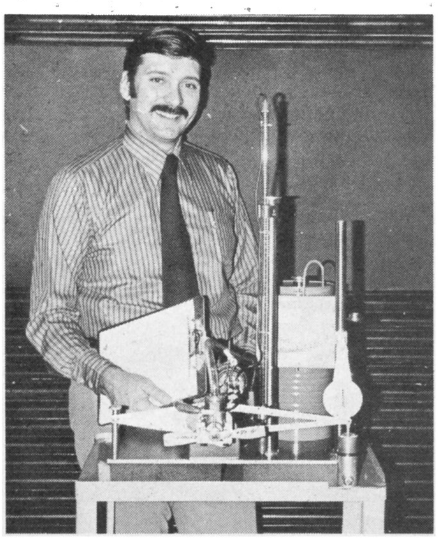
A. Conducting a dynamometer test.

B. Applying atmospheric correction factors.

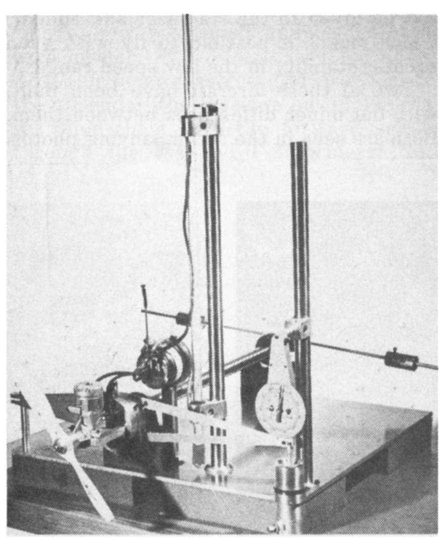
C. Analyzing dynamometer test results.

D. Energy flow through the 2-stroke cycle engine.

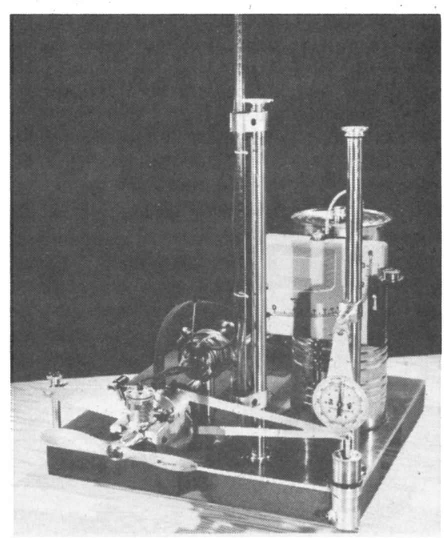
E. Air-fuel flow measuring system.



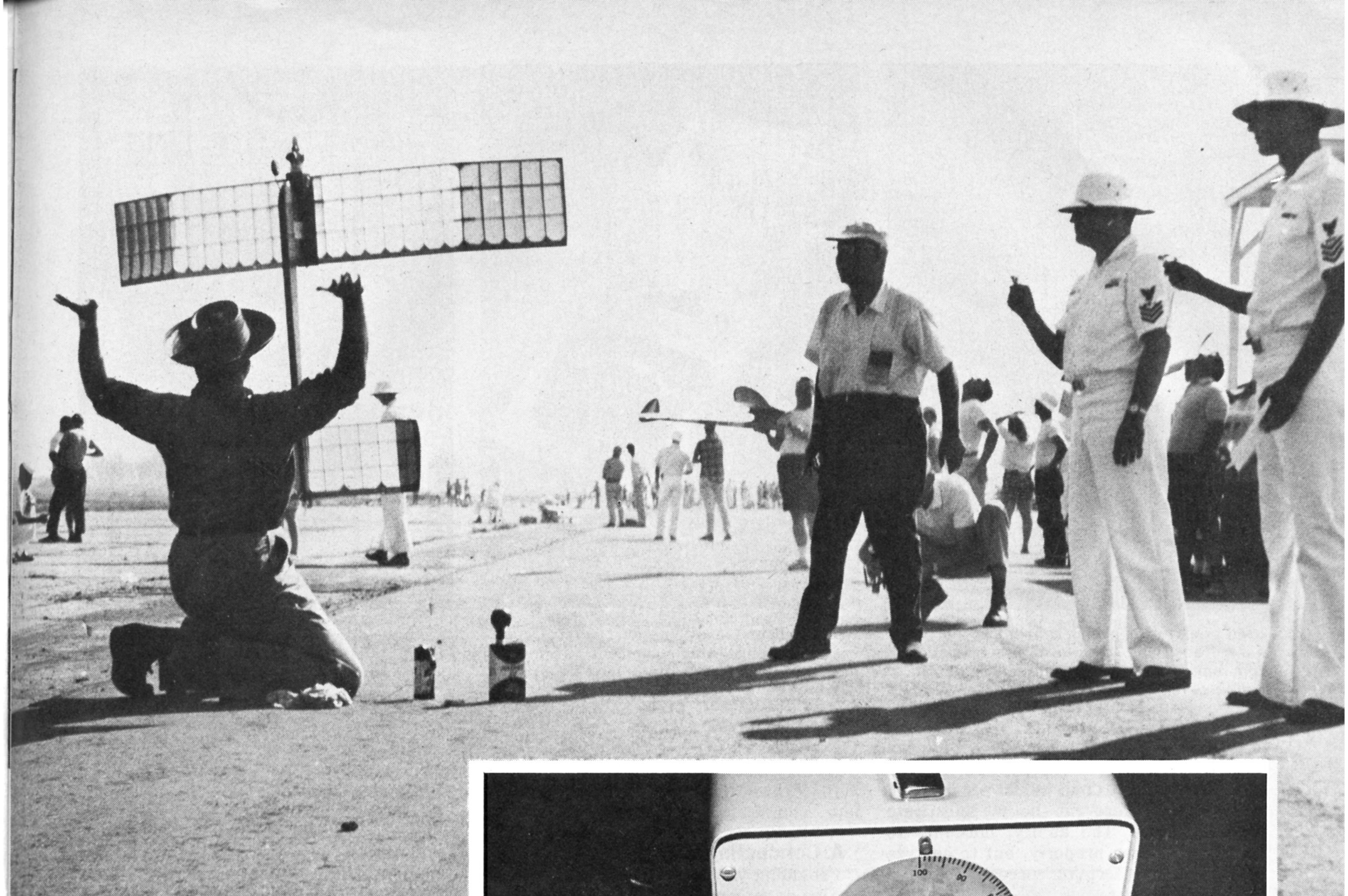
The word here is "that last head change got us another .10 at 21,000 rpm." Accurate measured results instead of just guessing from sounds.



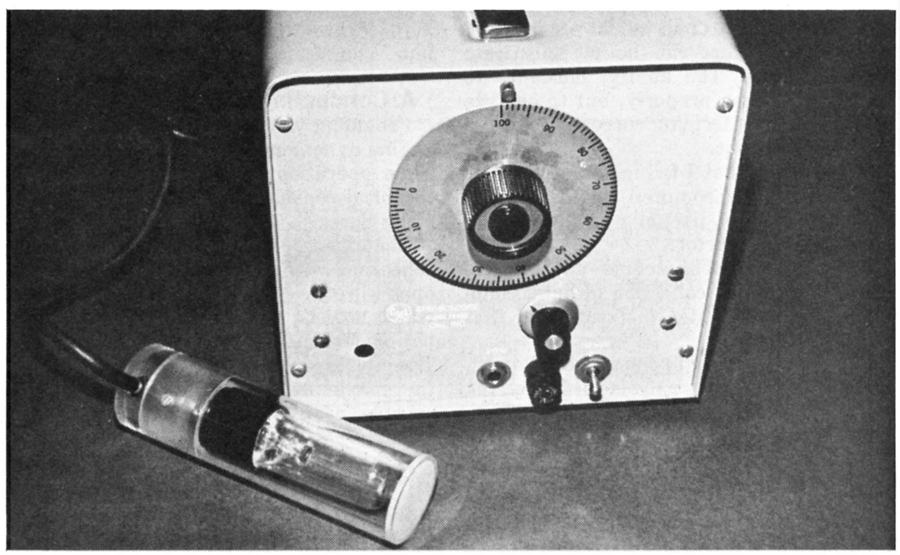
Engine analysis unit minus air flow measuring device with Super Tigre G-40 ABC strapped in the mounts. The end results were worthwhile.



Complete Dyno set-up with HP rear valve power. (The measure of a motor.) Interesting answers. Comparisons point way toward improvements.



A typical example of a hot powered Free-Flight that will benefit from Dyno testing and tuning. At right: White light Strobe records the rpm's.



Photos by the Author

## Part II

# DYNAMOMETER

## AN ENGINE PERFORMANCE ANALYSIS

**by Dave Gierke**

**Part II of our series explores the unknowns of powerplants. Dynamometer procedures, atmospheric factors, test results, the flow of energy through your engine. Intriguing reading!**

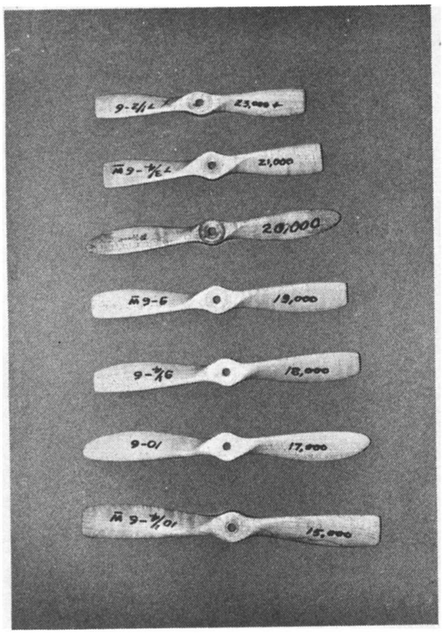
Hopefully, many of you survived the necessary "indoctrination" session from last month, where we attempted to "spell out" in specific language the concepts and principles dealing with energy systems analysis (Engine analysis).

You may find it helpful to refer back to last month's material for clarification purposes related to our future discussions.

Before "jumping in with both feet," I feel compelled to relate my feelings toward this controversial subject of engine performance. For many years, the "secret" of outstanding engine performance was seemingly limited to a few talented individuals who could be depended upon for record-setting performances. In discussing the "technology" of "how to make an engine go," a wide variety of rework procedures could be expected from the experts who were willing to talk. Others, of course, did not wish to divulge the "means" through which the "end" was accomplished. The only certain thing seemed to be confusion and the general lack of a common yardstick by which engine performance could be measured.

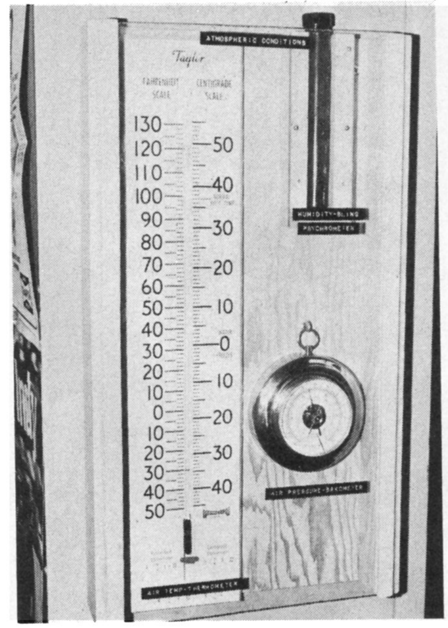
The "experience factor" has reigned supreme for as many years as I can remember. In the area of engine modification, for the purpose of performance increase, most of us are left out in the cold, as far as experience is concerned. A look at the top speed people in our hobby shows the obvious importance of experience in the area of engine modification.

Of course, the inexperienced modeler interested in racing or speed competition can purchase a reworked engine from one of the several good specialists working




Load props ranging from a high load (large) to a low load (small). Shown here is approximate rpm range for each. Blade finish is important.


Heath Thumb Tach is fine for field spot check, but more sophisticated equipment for Dyno rpm measurements is preferred for lab type results.




Atmospheric humidity sling Psychrometer, pressure Barometer and Thermometer. The correction factors neutralize variables in the testing.

in this field. However, how much time can this specialist afford to spend with your engine for the $20.00 to $40.00 rework fee?

Obviously, there can be no substitute for experience. The ability, not only to modify an engine properly, but to analyze *why* it is not performing correctly, certainly points this out.

Recognizing that I fell into the inexperienced category, prompted me to research the possibility of partially circumventing the experience factor.

As you shall see, engine analysis requires the *careful* collection of data, coupled with a systematic analyzation of that data.

In this issue, we will cover the following aspects of engine analysis utilizing the dynamometer and related instrumentation.

A—Procedure for conducting a dynamometer test.

B—Application of atmospheric correction factors for horsepower.

C—Analyzation of dynamometer test results.

D—Discussion of energy flow through the 2-stroke cycle engine.

E—Air and fuel flow measurement system.

## A. Conducting a Dynamometer Test

Providing you may build a simple, small engine dynamometer for testing purposes, some operational suggestions as well as technical considerations should be passed along here.

The dynamometer test which we are concerned with here is known as the "wide open throttle test." This is but one test which may be performed on the machine, but is the most useful for our purpose. The "wide open throttle" test shows us the maximum horsepower an engine is capable of delivering at the crankshaft (B.H.P.).

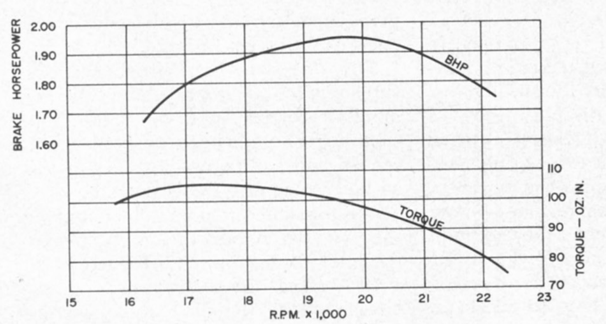
We will determine the maximum power of the engine at each R.P.M. level which is produced by the engine turning various sized loads in the form of propellers. The larger the diameter of the prop, the more "loaded down the engine becomes, producing a *lower* R.P.M. (See prop photo).

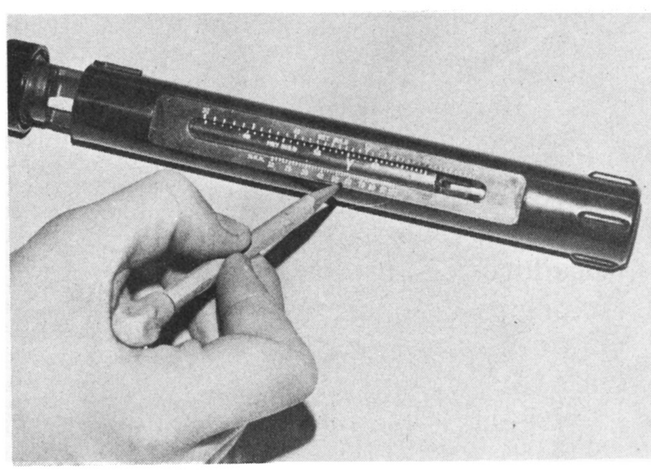
When testing an engine at wide open throttle (W.O.T.), you will discover that the horsepower is not the same at different R.P.M. A one horsepower engine, for example, may produce one horsepower at a certain R.P.M., but at all other R.P.M.'s, it will probably deliver less. (See Figure #1).

Because our racing engines in R/C pylon are not operated at the maximum rated horsepower R.P.M. at all times, it is just as important to know what the horsepower is at higher and lower R.P.M. In R/C pylon racing, the engine cannot maintain its maximum rated horsepower R.P.M. because the airplane slows down in turns resulting from increased drag, which in turn, induces an increased *load* on the prop, thus reducing engine R.P.M. Reducing engine R.P.M. creates a situation, which we will discuss later in which we are now operating below the maximum B.H.P. point on the curve. Torque, as we

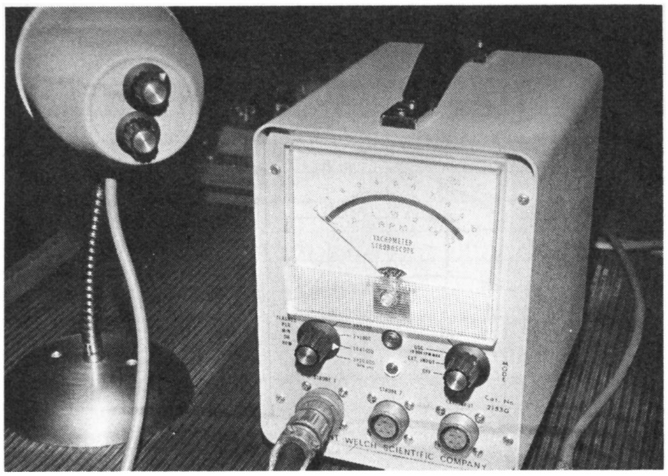
**Fig. 1**

| LOAD PROP | R.P.M. | TORQUE (OZ. IN.) | BHP |
|---|---|---|---|
| 10–6 | 16,300 | 103.0 | 1.67 |
| 9 1/2–6 | 17,000 | 105.0 | 1.79 |
| 9–6 | 19,500 | 100.0 | 1.95 |
| 8 1/2–6 | 20,000 | 97.0 | 1.94 |
| 7–6 | 21,800 | 83.0 | 1.80 |

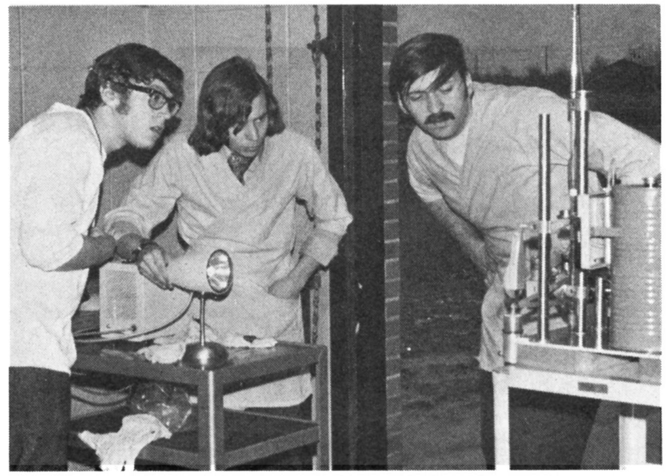


Wet bulb and dry bulb Thermometer being read. Humidity correction factor is determined by using a humidity chart. In the interest of performance.




An rpm measuring device. This white light Stroboscope is very accurate. Access to such exotic testing equipment has kept modelers in guesswork.



White light strobe here used to zero in on engine's rpm. Student teacher Tom Porebski is manipulating the controls for a reading on torque scale.

shall see later is an important factor along with the above-mentioned horsepower. Controline speed has considerably less problem, because the loads on the prop are not continually fluctuating. The engine may be propped to the maximum B.H.P. point without having to consider *acceleration* to maximum B.H.P. as in R/C pylon.

You will also observe that on all engines, the maximum torque will be developed at a *lower* R.P.M. than the maximum horsepower. (See Example Figure #1).

As shown in Figure 1, as a typical test curve for horsepower and torque versus R.P.M., I obtain data at five to seven different engine R.P.M.'s and then calculate the engine horsepower at each R.P.M. (Figure #2). A graph is constructed similar to the one shown in Figure #1 for the engine in test. For each R.P.M. load point, the engine must be shut down in order to fit the next size prop which will force the engine to run at another R.P.M.

As mentioned last month, a tachometer of high accuracy is required in order to obtain consistent R.P.M. readings. We have access to a strobe-o-scope which can be made to indicate R.P.M. through the use of a conversion chart from flashes per second to R.P.M. As seen in some of the photos, a Kavan mechanical tachometer is also highly recommended. We have found all of the light reflective, photo-electric cell type tachometers to be unsatisfactory, because of low accuracy.

Other critical factors to keep in mind while running a dynamometer test are the atmospheric conditions in which the test has been conducted.

## B. Atmospheric Correction Factors for B.H.P. (Corrected Horsepower)

The horsepower of an engine measured by a dynamometer is the actual horsepower under test conditions. However, *density* of air changes significantly with atmospheric temperature, pressure, and humidity. These changes affect the horsepower output of the engine. *Corrected horsepower* is the estimated horsepower of the engine under other atmospheric conditions. These "other atmospheric conditions" are known as standard temperature, pressure and humidity. These standards are given below:

Temperature—60°F.

Pressure—29.92 inches of mercury (sea level)

**Fig. 2**

DYNO B.H.P. FORMULA

$$BHP = \frac{F(OZ.) \times R.P.M.}{100{,}000}$$

EXAMPLE:

TEST LOAD – 15,000 R.P.M.

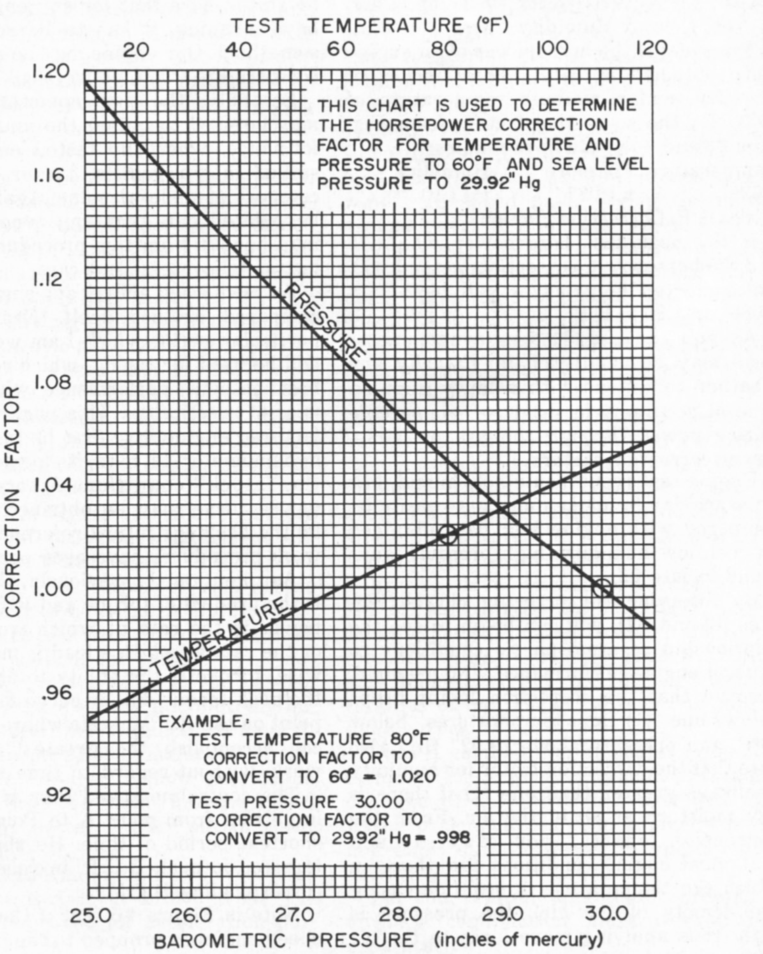
SCALE READING – 5 OZ.

$$BHP = \frac{5\ OZ. \times 15{,}000\ R.P.M.}{100{,}000}$$

$$BHP = \frac{75{,}000}{100{,}000}$$

BHP = .75 at 15,000 R.P.M.

**Fig. 3**



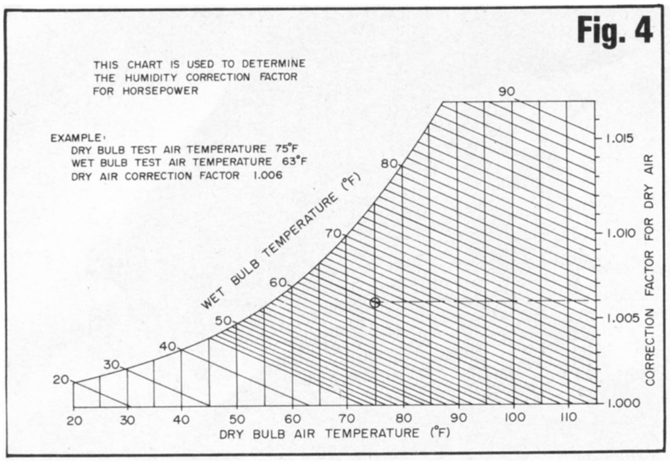



Fig. 4

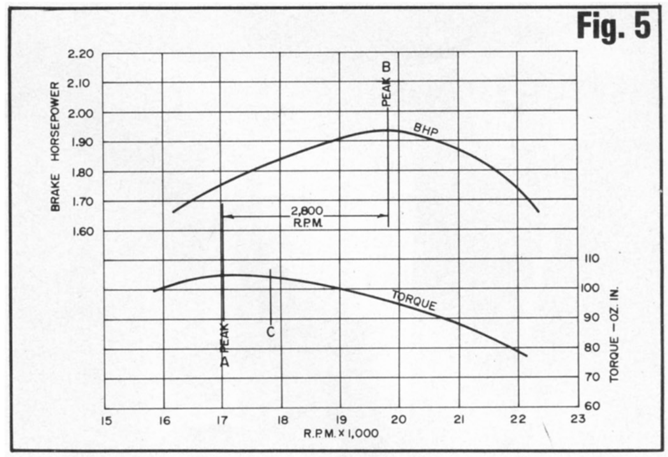



Fig. 5

Humidity — Dry Air (No moisture)

Because our engines are prone to great power changes depending upon the atmospheric conditions, it is imperative that each test be subjected to the horsepower correction procedure, which is designed to neutralize the atmospheric variable.

See Figure #3 for the chart to determine the horsepower correction factor for temperature and pressure to 60°F. and sea level pressure (29.92" Hq). Refer to Figure #4 for the chart to determine the horsepower correction factor for humidity. The horsepower correction factor is the result of multiplying the individual correction factors for temperature, pressure and humidity.

H.P. Correction Factor = Temp. Corr. x Pres. Corr. x Humidity Corr.

*Example:* If the horsepower of an engine were obtained at 85°F., 28.80 inches of mercury, and a wetbulb temperature of 75.00°F., the correction factors from Figures #3 and #4 would be as follows:

| Temperature | Pressure | Humidity |
|---|---|---|
| 1.039 | x 1.037 | x 1.010 = |

1.088 (H.P. Correction Factor)

If the peak test horsepower was 1.75 and the horsepower correction factor 1.088, the corrected horsepower will be 1.75 x 1.088, or 1.90 B.H.P.

As you can readily see, dynamometer tests may be conducted in virtually any weather conditions with a fairly high degree of accuracy, in terms of *comparable* engine power through the use of horsepower correction factors.

Engine test results on a hot, humid, low pressure day in midsummer may easily be compared with engine test data obtained on cool, low humidity, high pressure days found in late fall.

By observing the correction charts (Figures #3 and #4), you can begin to see the relationship of atmospheric conditions to general engine performance. For example, observe that the correction factor drops below one (1) for temperatures below 60°F. and pressure above 29.92" Hq. Also note that the correction factor for humidity is always greater than one (1) if there is any moisture at all in the air. From the correction charts, you may deduce that ideal engine conditions are those in which the temperature is low (increasing the density of the air), the pressure is high (thus applying more force to the air while pushing it into the engine), and low humidity (reducing the number of water molecules which would normally displace usable oxygen molecules in the air.

## C. Analyzation of Dynamometer Test Results

Now that we have some test data from the dynamometer and have graphically represented it in terms of R.P.M., Torque and Brake horsepower, what does it tell us?

This question probably can be divided into two parts. First, the data tells us something about the particular engine's performance in comparison to other engines. The "other" engines may be of the same make or of a competitor's brand. It should be stated here that for any engine test to have meaning, it has to be compared to something. Our engine may be successfully compared to itself, as we shall see.

Secondly, the dynamometer test data may be used to obtain the maximum performance, when the tested engine is installed into a vehicle, in our case, some category of model airplane. Again, it should be pointed out, that any type of vehicle may benefit from this procedure. Namely, boats, helicopters, and cars.

As mentioned above, our engine may be compared against itself. Example: The particular engine which I am working with will be used for racing, which requires the best possible performance within the limits of practicality. This particular manufactured engine may *first* be tested on the dynamometer to note its performance in the "stock" condition. A torque-horsepower curve may be obtained which will be the basis for future reference, after rework procedures have been performed.

As mentioned previously, horsepower is the product of torque and R.P.M. Horsepower is the rate at which work is done or the rate at which a load is moved. Likewise, torque is the ability to do work and the peak torque on our curve indicates the point on the R.P.M. scale where the engine can accomplish the greatest amount of work, without regard to time required.

The controline speed flier is interested in getting from point A to Point B in the shortest period of time. He should be interested in peak B.H.P. in relation to the corresponding R.P.M.

Note in Figure #6 that if the controline speed engine is propped to the peak B.H.P. point, it lies very low on the torque curve. This means two things: 1) In order to achieve the necessary R.P.M. to reach peak B.H.P., the size of the load (prop) will be reduced greatly. 2) The model's acceleration, because of the poor location on the torque curve, will be very low. Again, torque is the ability to do work, which indicates that with this low factor we also have a low acceleration rate. In the case of the controline speed model, acceleration accounts for very little in the final results. The low torque factor simply prolongs the model's ascent to its maximum velocity. One very notable exception to this general rule in C-L speed is the proto-speed model, which requires a high rate of acceleration from its standing start, as well as good top end velocity. This event, as with R/C pylon requires a prop compromise.

To a controline speed modeler, the torque-horsepower curves are useful from the aspect of peak B.H.P. Once he has performed his tests on the dynamometer for a given engine, he can then determine at what R.P.M. he must run his engine in the model in order to obtain this peak B.H.P.

Another point to consider is that the test figures from the dynamometer represent *static* running conditions. Under flight conditions, propeller R.P.M. will tend to increase due to a change in the propeller load. Propeller load will actually decrease when it is both rotating and traveling forward. To operate an engine at peak B.H.P.-R.P.M. under flight conditions, the propeller load would be selected to give a lower static R.P.M. But how much lower? This is an impossible question to answer accurately because vehicle drag is an important controlling variable. Trial and error technique may be employed here in determining the correct prop or another scientific procedure may be used.

In the case of the controline speed model it is a relatively simple procedure to determine what the in-air R.P.M. actually is. While the model is being flown with the engine in question, another person is stationed in the circle with the pilot reading engine R.P.M.'s with the use of an audio-tachometer. The audio-tachometer utilizes the operator to match the "beat" set up between the engine's exhaust note frequency and the actual tachometer which generates its own audio frequency. The tachometer is calibrated to read-out, in R.P.M. and is generally quite accurate.

Fig. 6

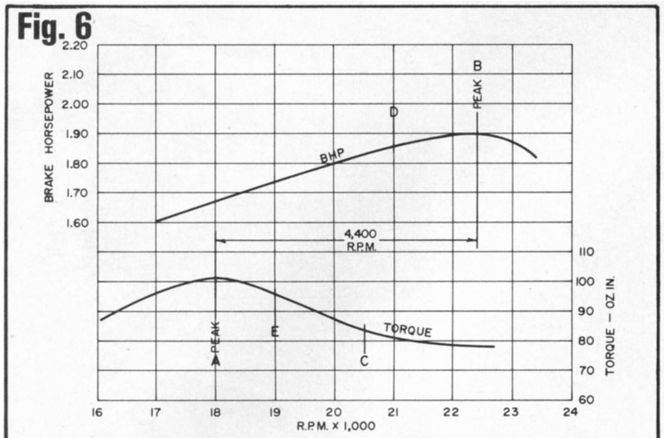
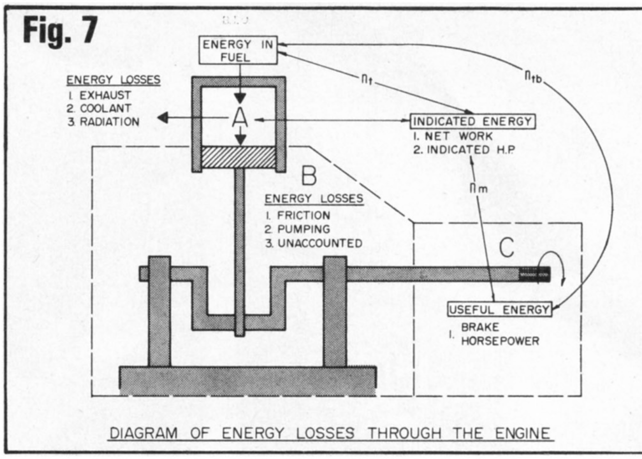



Fig. 7




The tachometer operator must be in the center of the circle with the flier in order to eliminate "Dopler effect" which otherwise makes accurate readings difficult to obtain.

Whereas the controline speed model is concerned primarily with B.H.P., the R/C pylon racer is involved with several other factors which make it quite a bit more difficult to obtain satisfactory results.

As prominent East coast pylon racer, Jerry Wagner, suggests, "The pylon course is like a drag strip—you accelerate to one end, turn, decelerate, and accelerate back" It might be added that during this "drag race", it is desired to have the maximum obtainable velocity (B.H.P.) possible. Unfortunately with a fixed pitch wooden propeller, as with the proto speed controline model, a compromise must be sought. To illustrate the above problem, please refer to charts shown in Figures #5 and #6.

In Figure #5 a typical torque-horsepower curve for a popular racing .40 is graphically represented. First, notice the rated values for B.H.P. and torque.

B.P.H.—1.94 at 19,800 R.P.M.

Torque — 105 Oz. In. at 17,000 R.P.M.

Letter A represents the torque peak while letter B indicates the B.H.P. peak. In this particular dynamometer test, the spread on the R.P.M. scale between A and B is 2,800 R.P.M. Likewise, it may be assumed that this is the spread between the maximum acceleration factor (torque peak-A) and the maximum velocity factor (B.H.P. peak-B). As mentioned, in pylon racing, we need acceleration from the high engine load turns and also the maximum straight-away velocity.

For purposes of comparison, observe the curve represented in Figure #6 of another popular brand of racing .40.

B.H.P. — 1.90 at 22,500 R.P.M.

Torque — 102 Oz. In. at 18,000 R.P.M.

In this dynamometer test, the maximum B.H.P. and torque *values* are very similar to those shown in Figure #6. Note however, that the R.P.M. spread between A and B (4,400 R.P.M.) is much larger than with Figure #5. Obviously, this is because the second engine does not develop its maximum B.H.P. until a much higher R.P.M. Thus, the greater spread between A and B.

Now, if in Figure #5 we prop this engine in our pylon racer to turn "in-air" R.P.M.'s of 19,800, the model will be operating at its maximum velocity in straight flight. In the actual race, however, the model must turn and fly a closed circuit course. And, as discussed before, the model will slow down in turns, because of increased drag. The engine will "load up" thus reducing its R.P.M.'s. Depending upon the propeller used, and the drag characteristics of the model being flown, the engine might slow down as much as 2,000 R.P.M. or more. In the case of the engine in Figure #5, this R.P.M. loss would be about ideal for maximum (letter C) acceleration capability. With a 2,000 R.P.M. loss, this will reduce engine R.P.M. to 17,800, which is virtually at the peak of the acceleration (maximum) point. Now accelerating, the model with increased velocity and decreased drag, will allow the engine to "unload" and move rapidly up its B.H.P. curve to the maximum point. This, of course, is ideal and is difficult to achieve in reality. The difficulty arises in finding the correct propeller and also the problem of engine power changing from day to day due to atmospheric conditions.

Now observe the curve in Figure #6 of the second popular racing .40. If this engine were propped to operate at 22,500 "in air" R.P.M.'s, it also would be operating at its maximum velocity in straight flight. If we apply the same R.P.M. loss due to the high drag condition of the model, in a turn, we would have an R.P.M. of about 20,500. Notice where this R.P.M. would locate itself (letter C) on the torque curve. The torque value for useful acceleration purposes is only about 84 oz. in. Compare this torque value with the one for Figure #5 which was about 104 oz. in.

The engine in Figure #6 will undoubtedly be slow in accelerating out of a turn and will probably never reach its peak B.H.P. point on the R.P.M. scale. Engine A from Figure #5 will appear to have much more power, but as you can see from the charts, brake horsepower is about the same.

A propeller compromise in the case of engine B in Figure #6 will undoubtedly improve its overall performance. By operating the engine at 21,000 maximum "in air" R.P.M.'s, the loss due to high drag conditions would drop the engine into the 19,000 R.P.M. range (letter E) for "out of turn" acceleration.

Note that the maximum *effective* peak B.H.P. point is 1.85 at 21,000 while the torque has improved for acceleration purposes to 96 oz. in.

The second engine (Figure #6) is a comparative short stroke engine which would be better suited for controline speed. A few people have been able to make this engine work by arriving at a narrow range combination of low drag airplane, correct compromise in propeller, and careful tuning.

There are three racing .40's available which are of the longer stroke variety and are similar in power to that shown in Figure #5. The close R.P.M. peaks between torque and B.H.P. can easily be seen. The longer stroke has its obvious advantage in pylon racing.

The question of determining what the "in air" R.P.M.'s are for a pylon racer are understandably more complicated than they were for the controline speed model. The dynamometer will tell you very little about "in flight" props as will any other *static* measuring device, because of the unloading condition of the prop while moving through the air.

Using the audio tachometer presents some difficult problems with "dopler" effect, while making an upwind and downwind high speed pass. Flying the model around yourself sounds like it may work, but in reality the engine "loads down" due to increased airframe drag. Several individuals have tape recorded the engine's sound while making an upwind and downwind pass. This recording was analyzed electronically, thus producing an average "in air" R.P.M.

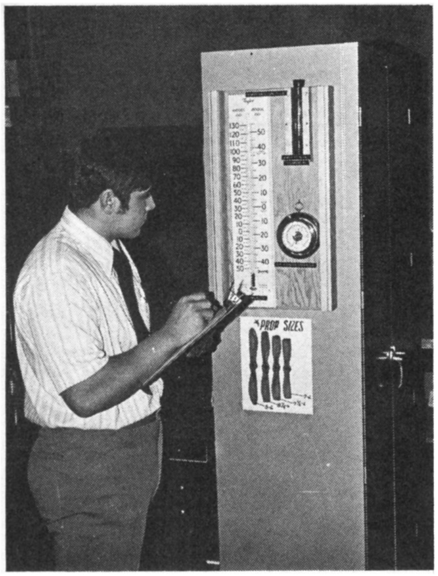
The most promising method seems to be the recently developed and described "telemetering" device which rides in the airplane. The device transmits back to ground all wanted data from onboard instrumentation. R.P.M. is one piece of data which would be invaluable.

As we have discussed, straight-away R.P.M. and turning R.P.M. are of importance while attempting to match the dynamometer data. Changing propeller pitch and diameters vary the "in air" characteristics of our model, in order to match dyno curves.

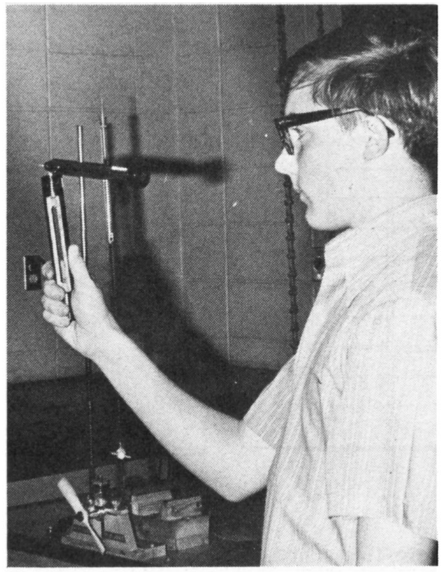
Once a combination of engine/propeller is found for a given airplane, it usually can't be used in another model with differing drag qualities. The engine will act noticeably different for each type of model used.

For this reason, each engine/propeller/airplane combination must worked out separately.

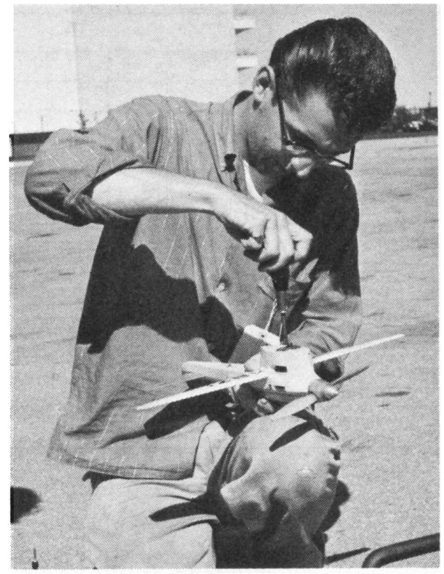
Analyzing engine performance during



Atmospheric conditions are big factor. Student teacher Joe Frontera busy recording specifics.



Student Al Kerner is slinging the Psychrometer for a humidity measurement. It influences rpm's.



Controline Speed ships could also benefit from Dyno analysis to achieve maximum performance.

various stages of a rework procedure is another useful application of the dynamometer. By changing engine specifications such as: rotor timing, sleeve (port) timing, compression ratios, scavenging methods, base compression, cylinder head shapes, etc., the B.H.P. and torque curves are understandably modified.

Maximum torque and B.H.P. peaks can be shifted and spread out or compressed on the R.P.M. scale. In some cases, the B.H.P. peak can be made to crest at better than 24,000 R.P.M. which is beyond the practical range of usability in R-C pylon. Although the maximum B.H.P. figure is impressive at this high R.P.M., it cannot be used.

The method most modelers use for lack of a better system, is the standard "test prop" comparison of one engine to another. For the racing .40 a 9-6 Super M has become almost standard. The test prop theory gives a good general comparison of maximum horsepower outputs, but tells us nothing about the location of critical torque and B.H.P. peaks.

*Note:* The following sections (D and E) are designed to give you some necessary background for next month's installment.

## D. Energy Flow Through the 2-Stroke Cycle Engine

Prior to our discussion of further dynamometer testing, it seems to be about the right time to take a "bird's eye" view of the energy flow through our engine. Figure #7 is schematic diagram of this flow.

The fuel is provided to the combustion chamber (A) where it is burned, converting *chemical energy into heat.* It would be desired to have all the liberated (thermo energy) energy drive to the piston, but there are heat losses, primarily due to the exhaust and to the coolant (air, in our case). The remaining energy, which may be converted to *indicated horsepower* (ihp), is utilized to drive the piston. The ratio of this energy to the energy originally supplied in the fuel, in comparable units, is termed *indicated thermo efficiency* (nt).

The energy applied to the piston passes through the connecting rod and crankshaft (Area B), to the driveshaft. Again, there are energy losses due to *friction, pumping* and other causes—the sum of all these losses, converted to power, is termed *friction horsepower* (fhp). The remaining energy (delivered to Area C) is that which can be utilized mechanically. This energy may be converted to power, and is termed *brake horsepower* (b.h.p.). The ratio of b.h.p. (delivered power) to i.h.p. (power provided to the piston) is called *mechanical efficiency* (nm).

The ratio of delivered energy (b.h.p.), in comparable units, to the energy originally provided by the burning of fuel is known as the *brake thermo efficiency* (ntb) of the engine. The brake thermo efficiency equals the product of the indicated thermo efficiency (nt) and the mechanical efficiency (nm). Also, the i.h.p. developed at the piston, minus the f.h.p. lost in the engine, equals the b.h.p. delivered at the driveshaft of the engine.

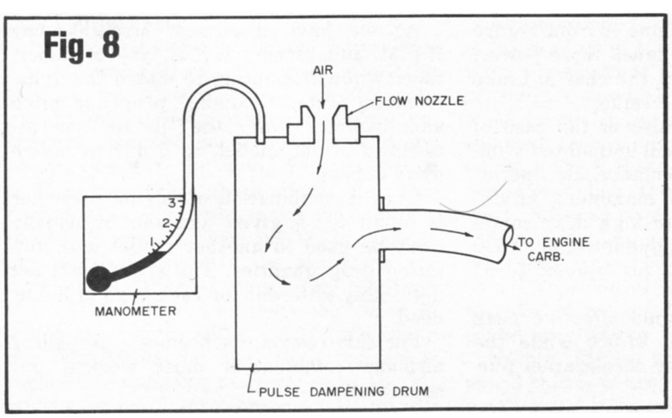
Above, it was mentioned that fuel is provided to the combustion chamber and burned to produce heat. While on the subject of the fuel and air supply, a brief mention of some of the associated parameters should be made.

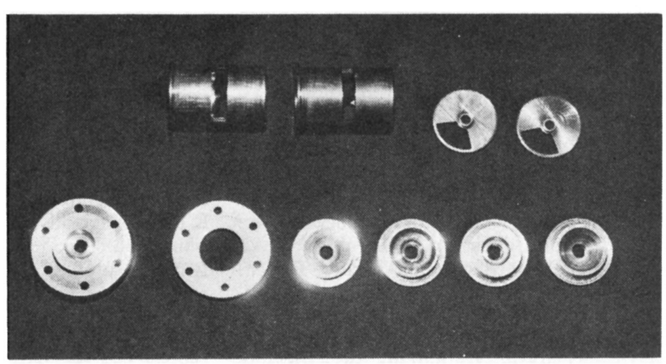
Energy supplied to the engine is in the form of chemical energy contained in the fuel. If the fuel is burned in the presence of oxygen in the atmosphere, heat energy is liberated. The amount of heat which one pound of a fuel is capable of liberating when it burns in the presence of oxygen is termed the *heating value* (H.V.) of the fuel, and is usually expressed in *British Thermo Units* (b.t.u.) *per pound of fuel.* This H.V. can be determined experimentally in the lab, for any given fuel.

## E. Air and Fuel Flow Rate Measurement

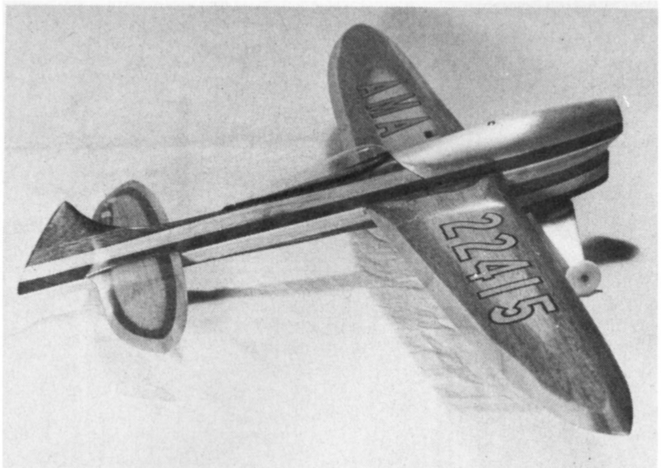
The four factors necessary for complete engine analysis are *R.P.M., Torque, Air and Fuel Flow Rates.* To this point we

Fig. 8



Our typical engine re-work procedures strive to squeeze out maximum bhp. Cylinder port timing, rotary disc induction timing, head configurations.




A typical Proto speed design minus an engine, prop and spinner. This is difficult C/L event as it combines great acceleration with top speed.

1972 National FAI Pylon champs. Garry Korpi and Luke Roy preparing to fire-up. The proper compromise of engine and prop must be found to win.

have discussed the measurement of R.P.M. and torque with its related calculations.

The B.H.P. output of our engine depends upon the quantity of energy liberated during the combustion of fuel and the air. The volume of the air utilized by the engine is many times the volume of fuel used. Since the useful air-fuel mixture range is restricted, the output of the engine is pretty much limited by the amount of air which can be inducted.

One of the most important factors concerning the B.H.P. output of the engine, therefore, is the induction of the greatest possible amount of air. More air induction permits the useful addition of more fuel, thus increasing the energy available to produce work. By measuring the exact amounts of *air* and *fuel* being consumed by the engine with an *air-flow meter* and a *fuel-flow meter*, a great deal more can be learned. After the air-fuel data has been obtained, it is possible to calculate:

*Volumetric Efficiency* (scavenge efficicncy-2 stroke)

Other factors may also be obtained from the measurement of air-fuel, flow rates, such as *Air-Fuel Ratio, Specific fuel consumption, brake mean effective pressure and brake thermo efficiency,* however, for our purposes *volumetric efficiency* is the most important.

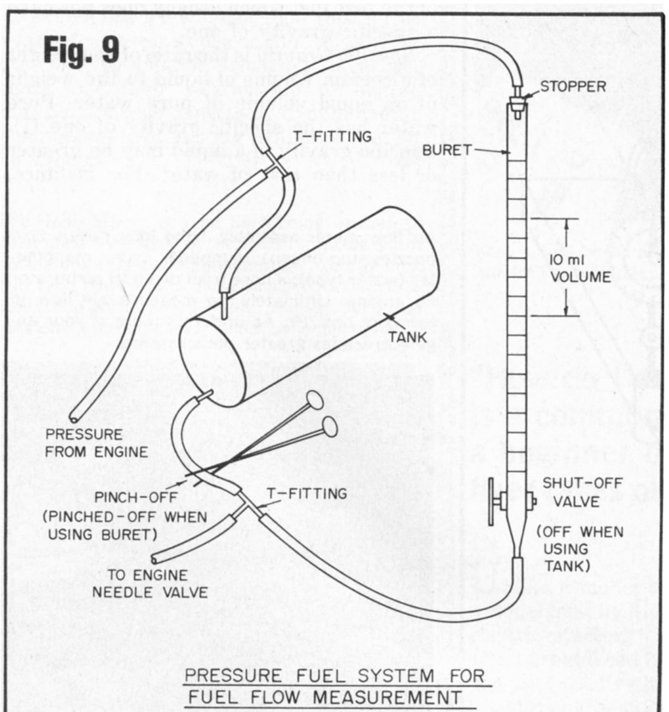
## Air Flow Measurement System

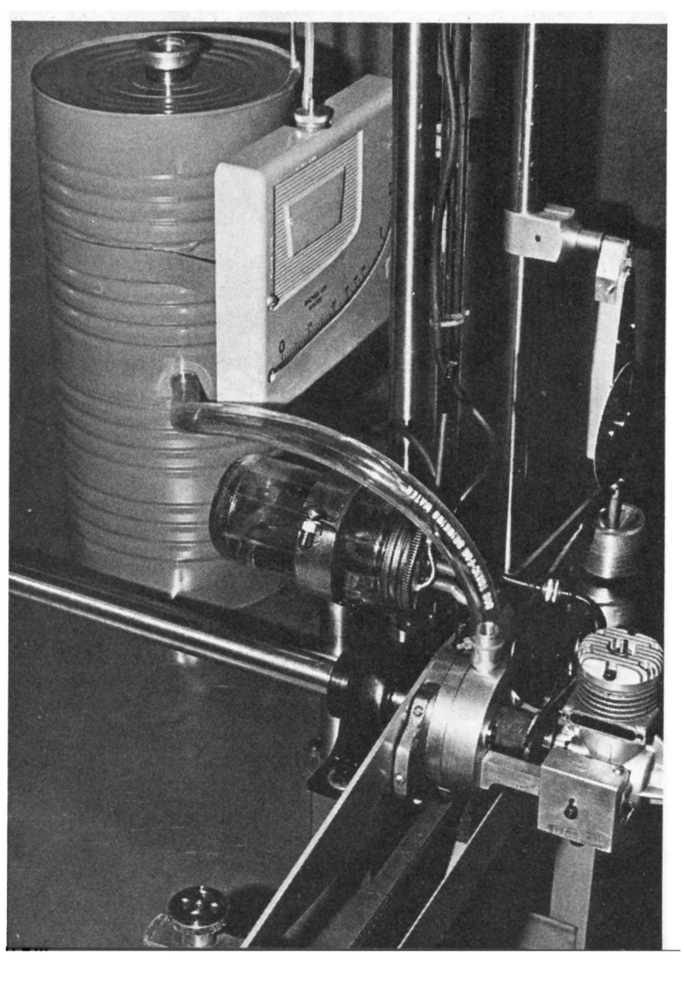
Air Flow is measured by drawing engine air through a precision flow nozzle into a *pulse dampening drum,* and then out through a flexible hose into the engine carburetor.

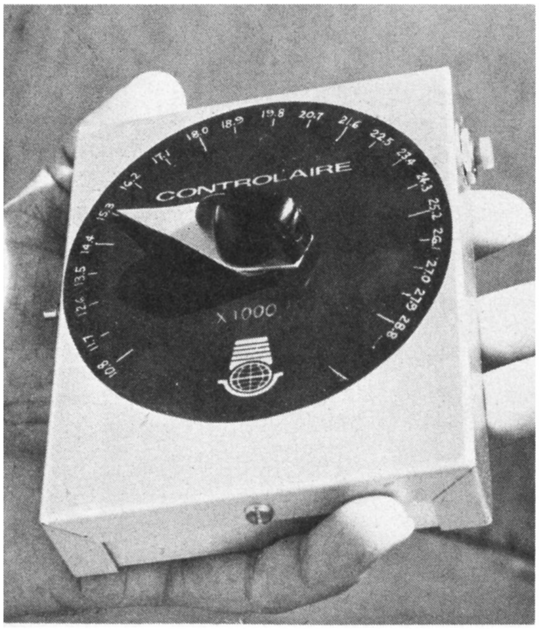
All of the air entering the carburetor must be drawn in through the nozzle. By measuring the pressure drop across the flow nozzle, the air flow rate can be calculated to a high degree of accuracy. The pressure difference is measured in *inches of water* by a pressure measuring device called a *manometer* (See Figure #8).

Record the inches of water pressure difference across the nozzle, in order to determine the air flow into the engine at each test R.P.M. (each load). The air-flow chart is then utilized to determine the air flow rate in *pounds/hour* (see next month).

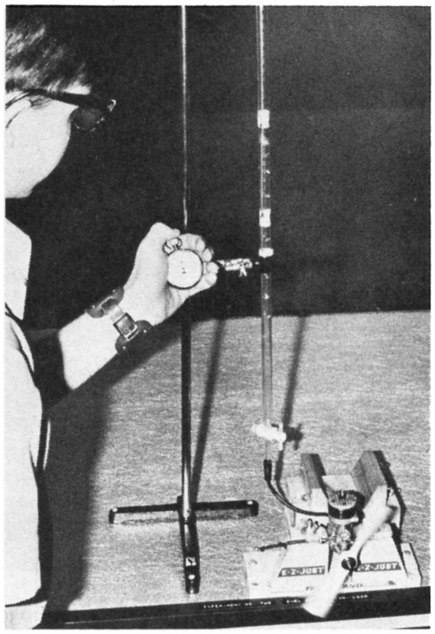
At right: It'll never fly! But it will help you fly better. Shown here, the Air and Fuel systems neatly connected to the basic Torque measuring instrument. I get the feeling that you learn something in Dave's class.



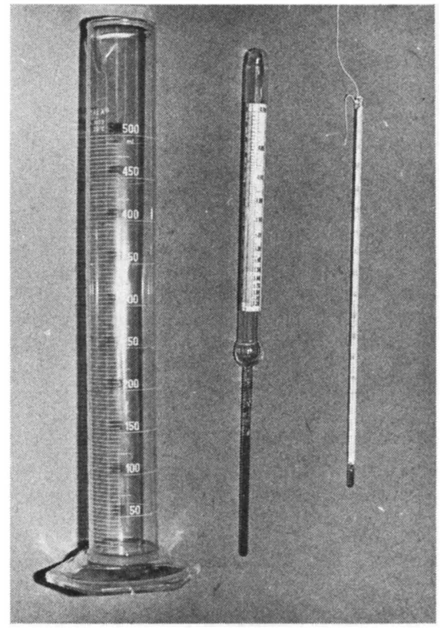





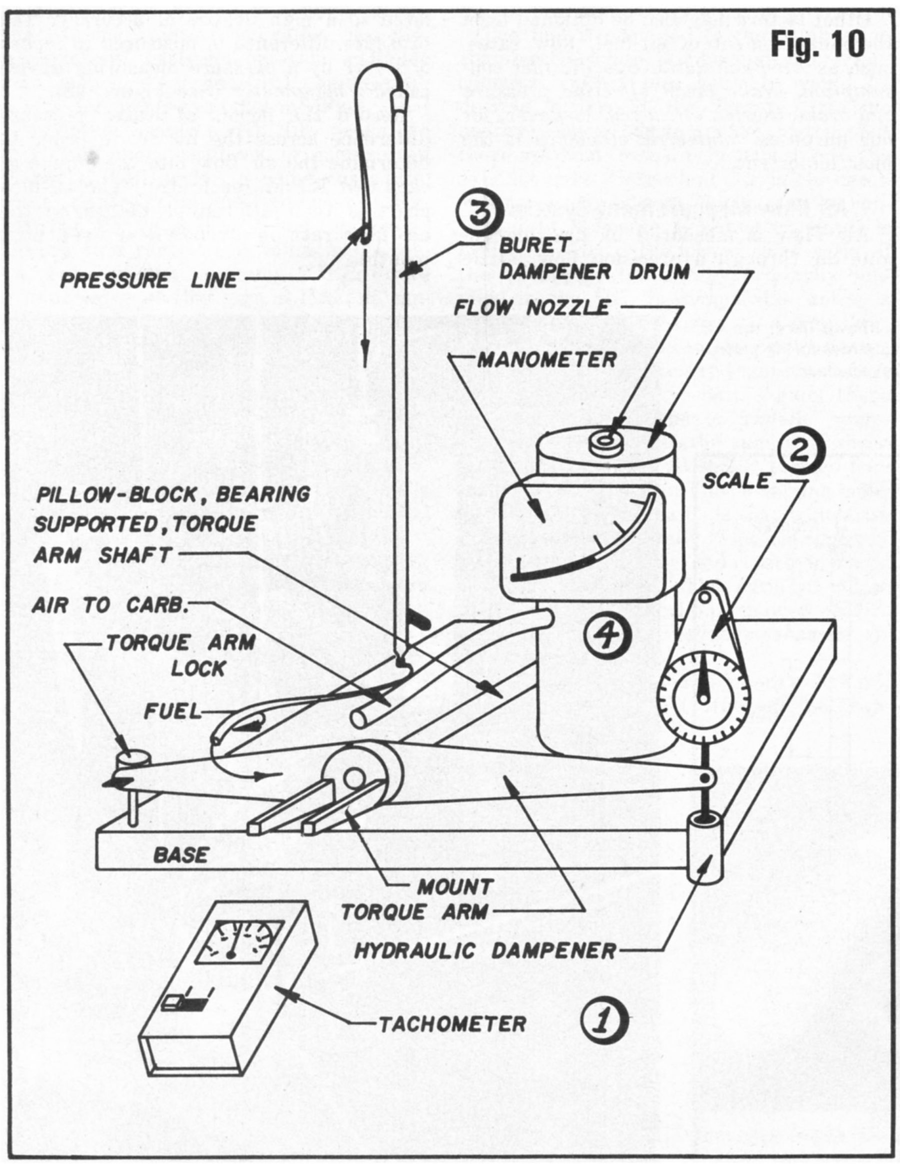

This is an Audio Tachometer. It's particularly useful in measuring rpm in Controline Speed. A recorded answer is far better than guesswork.



Al Kerner performing a basic fuel consumption experiment using the buret and stop watch. Note the old K&B .35 test engine. A good workhorse.



Left to right: 500 ml. graduated cylinder; a Hydrometer, Thermometer. Used to determine the specific gravity of our various fuel mixtures.




## Fuel Flow Measurement System

Fuel flow is measured by switching from tank to measuring *Buret* during the engine test, at each R.P.M. (load).

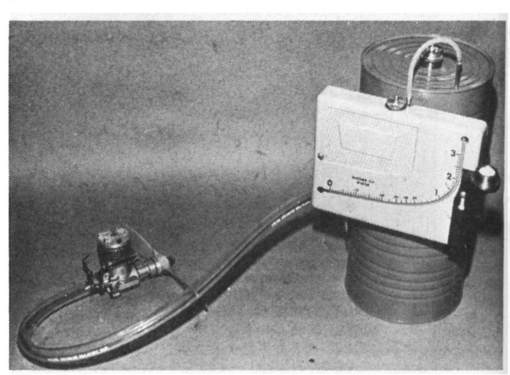
The Buret holds 50 ml of fuel but only a 10 ml segment is used for measurement purposes. As the fuel level drops rapidly in the buret, a stop watch is used to record the time in seconds required for the engine to consume the 10 ml volume (Figure #9).

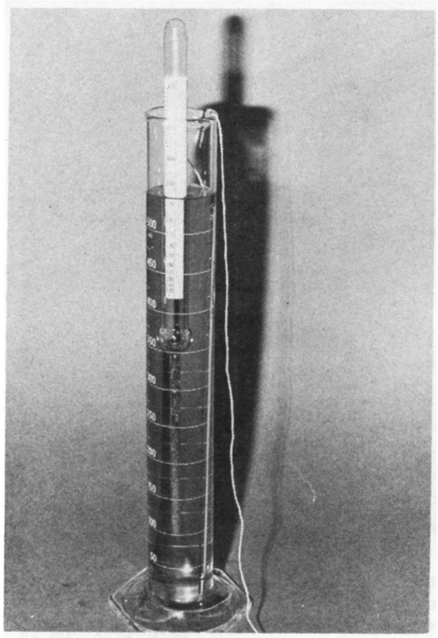
The fuel-flow chart is then utilized to determine the fuel flow rate in *pounds per hour* (discussed next month).

The *fuel-flow chart* has been calibrated to a *specific gravity* of one (1) which requires the correction in pounds per hour of the test fuel which usually does not have a specific gravity of one.

*Specific Gravity* is the ratio of the weight of a certain volume of liquid to the weight of an equal volume of pure water. Pure water has the specific gravity of one (1). Specific gravity of a liquid may be greater or less than that of water. For instance,

Air flow meter assembly. Note long radius flow nozzles (top of can); dampener drum; manometer (water type); a hose from drum to carburetor on engine. Ultimately we measure air flow in terms of Lbs./Hr. An understanding of your engine precedes greater performance.





A Hydrometer is being used to directly measure specific gravity of fuel. Temperature must be 60 degrees to control density as the standard.

the specific gravity of the acid solution in an ordinary storage battery is about 1.28 when the battery is fully charged. Depending upon the fuel used, the specific gravity could be both lower or higher than one (1). i.e.-nitro methane sp.gr.—1.14; methanol sp.gr.—.8

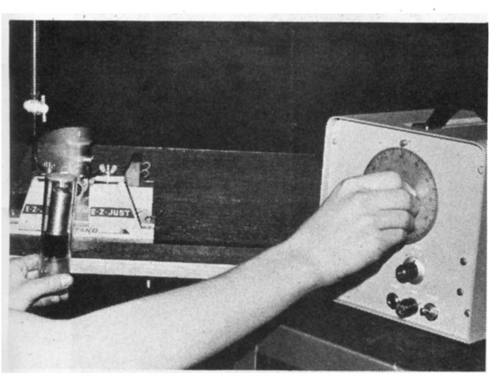
The specific gravity for our test fuel may be quickly found by using a floating *hydrometer* which may be purchased at a local scientific supply house.

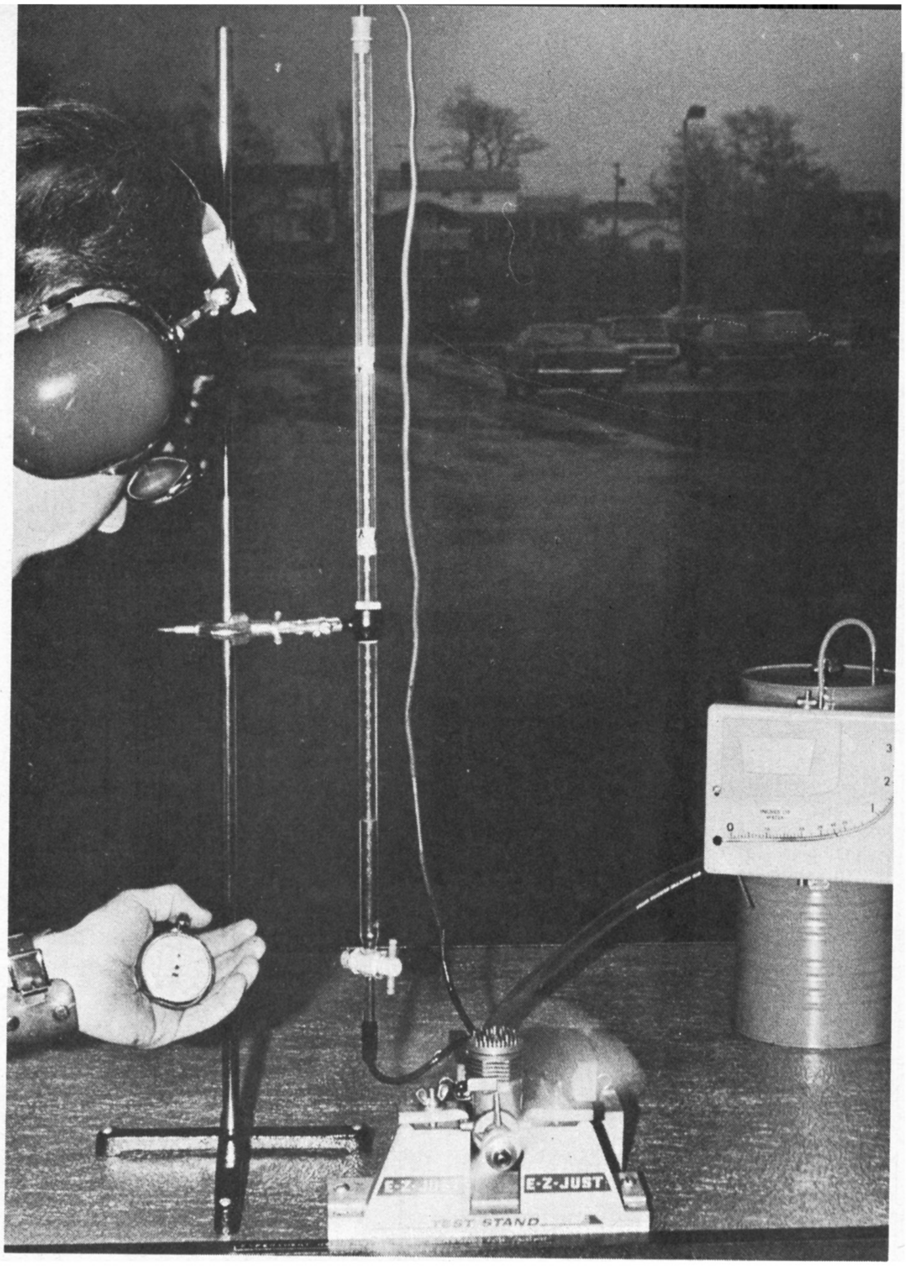
Example: If our fuel was found to have a specific gravity of .85 (K&B 100) and the measured fuel flow rate was found to be 6 pounds per hour, the correction would be as follows:

(Six) 6 lbs./Hr. × .85 specific gravity of test fuel = 5.10 Lb./Hr. Actual

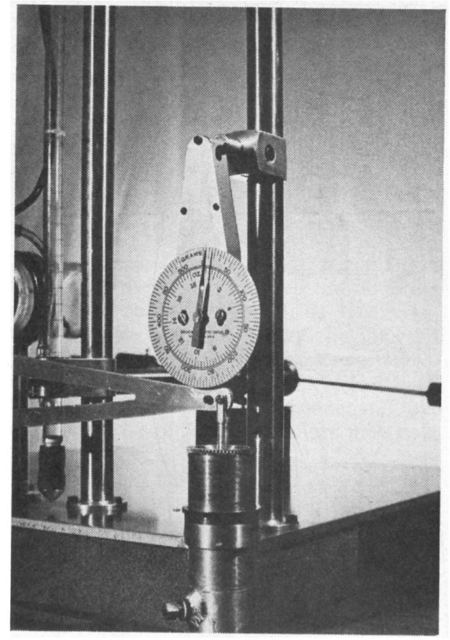
Next month we will conclude this series of articles with a discussion of *air-fuel ratio* and *volumetric efficiency* measurements and their relationship to engine analysis. Various aspects of dynamometer construction and testing will also be on the agenda.

The Neon Strobe is seen here being utilized to determine the rpm's of the K&B .35 powerplant during one of the fuel consumption experiments. Lab type results are more easily obtained in a controlled atmosphere, rather than under great tensions and stresses found at a flying field.






Student Al Kerner performing an Air/Fuel consumption experiment using the Buret/stop watch for fuel and Monometer/flow nozzle/dampener drum for the air. Repeat experiments to average things out.



Part III

# DYNAMOMETER

## AN ENGINE PERFORMANCE ANALYSIS

by Dave Gierke

**On scavenge efficiency, air to fuel ratios, construction of instrumentation, analyzing results and other meaty data to wind up this in-depth study of your engine's performance.**

From last month, you will recall that we discussed some procedures for conducting a dynamometer test where *torque* was our main concern. We also considered a method of neutralizing the effects of atmospheric conditions on engine performance for dyno tests by using B.H.P. correction factors.

The analyzation of dynamometer tests in relation to engine and vehicle performance was of prime concern also. Finally, energy flow through the engine and methods of accurately measuring it (air-fuel) were contemplated.

In this concluding issue, we will discuss the following aspects of engine performance analysis:

A. *Volumetric Efficiency (Scavenge Efficiency)*
   1. Formula
   2. Air Flow Chart
   3. Air Flow Correction Charts
      a) Temperature and pressure
      b) Humidity
   4. Example
   5. Controlling factors

B. *Air—Fuel (A/F) Ratio*
   1. Formula
   2. Stoichiometric; "rich"; "lean"
   3. Fuel Flow Chart

C. *Other Calculations*
   1. Specific Fuel Consumption (SFC)
   2. Brake Thermo Efficiency ($\eta$tb)
   3. Brake-Mean-Effective-Pressure (BMEP)

D. *Conducting a Dynamometer Test*
   1. Engine Analysis Data Sheet
   2. Test Procedure
   3. Test Data Graph

E. *Analysis of Air/Fuel Flow*
   1. Scavenge Efficiency—Air Charge
   2. Air Consumption
   3. Air/Fuel Ratios

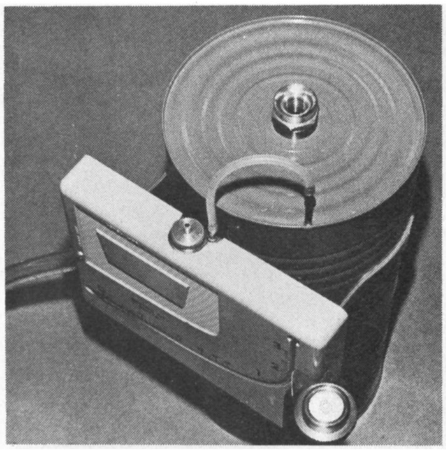
F. *Other Specific Dynamometer Tests*

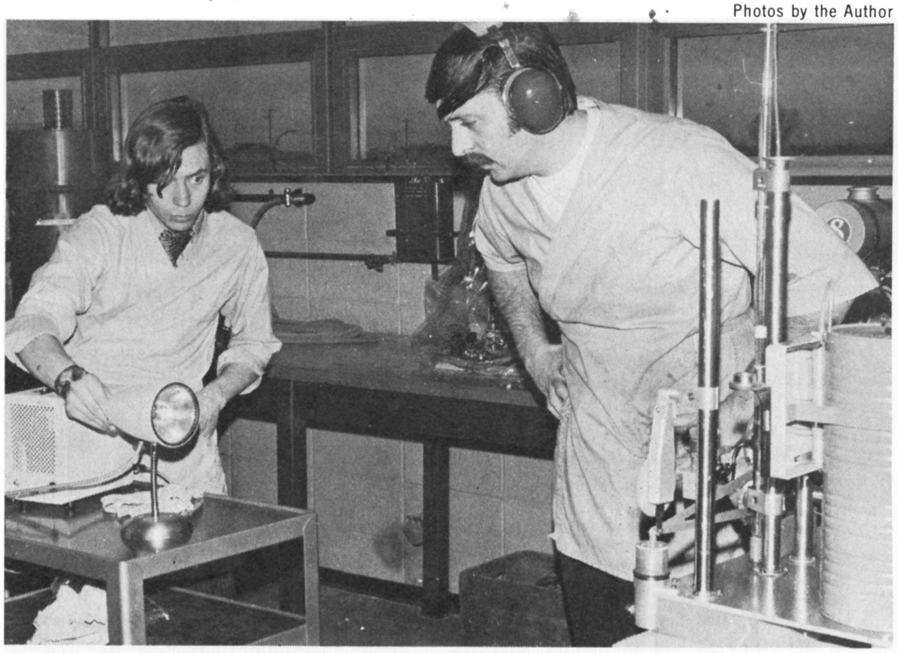
G. *Instrumentation Construction*
   1. Dynamometer
   2. Air Flow Unit

### A. Volumetric Efficiency

In two stroke cycle engine operation, the term "volumetric efficiency" is some-

Photos by the Author





Flow Meter, showing the use of the flow nozzle through which all air has to enter for engine. At left: Torque measuring device spring scale. Right: RPM's measured by a white light strobe. The knowledge of what is happening within the roaring engines has eluded us for decades, it is difficult to record by the normal means of research, testing and evaluation of data.

what meaningless, because the engine requirements for air are different from those of the four stroke cycle engine. In this case, in order to produce good scavenging, some of the air in the fresh charge is lost through the exhaust ports. As a result, the definition of volumetric efficiency is usually not applied to this type of engine. A wide variety of terms such as *scavenging efficiency, delivery ratio* and others, which express generally the ability of the engine to utilize through combustion the maximum amount of air supplied. Stated another way, we might say, power output from a two-stroke cycle engine depends principally on how well it breathes. Very important is the delivery ratio (scavenge efficiency). That is: The mass of each fresh charge, expressed as a ratio to the mass needed to fill the piston displacement volume at standard atmospheric conditions (temperature 60° F; pressure—29.92 inches of merc.).

A large air delivery ensures high B.H.P. if the charge remains in the cylinder and is not lost through the exhaust ports. Large delivery and entrapment of the charge depends on the detailed design of the internal engine layout. Later we will discuss some of these design features which affect scavenge efficiency. It should be added that the intake and exhaust systems play a large part here if they can be utilized.

## Formula

$$\eta r = \frac{\text{W act.} \dfrac{(\text{Lb. air} \quad )}{(\text{Unit Time})}}{\text{W theor.} \dfrac{(\text{Lb. air} \quad )}{(\text{Unit Time})}}$$

$\eta r$=Volumetric effiency (Scavenge Eff.)
W act.=Actual Weight Rate
W theor.=Theoretical Weight Rate

The equation for calculating volumetric efficiency in a *2 stroke cycle* engine is:

Volumetric (Scavenge) Effic. (%)=

$$37{,}750 \times \frac{\text{corrected Air Flow (Lb. Hr.)}}{\text{Engine in}^3 \times \text{R.P.M.}}$$

Volumetric (scavenge) efficiency changes with engine R.P.M. The air flow rates are recorded for each load which is placed on the engine.

See—*Air Flow Chart*—Figure #1
See—*Air Flow Correction*
Temperature and Pressure—Figure #2a
Humidity—Figure #2b

The figures of the air-flow chart (Figure #1) are for dry air at 60° F. and 29.92 inches of mercury (barometric pressure at sea level). This is standard temperature and pressure (STP) as was discussed earlier for B.H.P. correction factors. The tests we will conduct probably will not be at S.T.P. and the air will contain some moisture; therefore, it is necessary to *correct the air-flow* rate for the actual temperature, barometric pressure, and humidity of the air at the time of measurement. Without correction, the air-flow data on the chart could be in error by as much as 30 per cent.

The correction factors can easily be obtained from the correction charts (Figure #2a and 2b).

The correction factor is the product of the correction factors for temperature, pressure and humidity as determined from these charts.

### Example

If the test had been conducted at 95° F. and 28.50 inches of mercury (barometric pressure) and a wet bulb temperature reading of 75° F., the correction factors from figures 2a and 2b would be as follows:

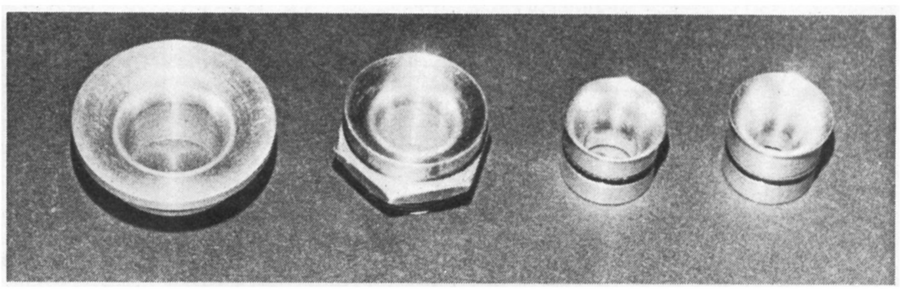
$$\frac{\text{Temperature}}{.954} \times \frac{\text{pressure}}{.926} \times \frac{\text{humidity}}{.978} = 0.864$$

$$\frac{\text{Measured Air Flow (Lbs/Hr.)}}{15 \times 0.864} = \frac{\text{Corrected Air Flow (Lbs/Hr.)}}{12.96}$$

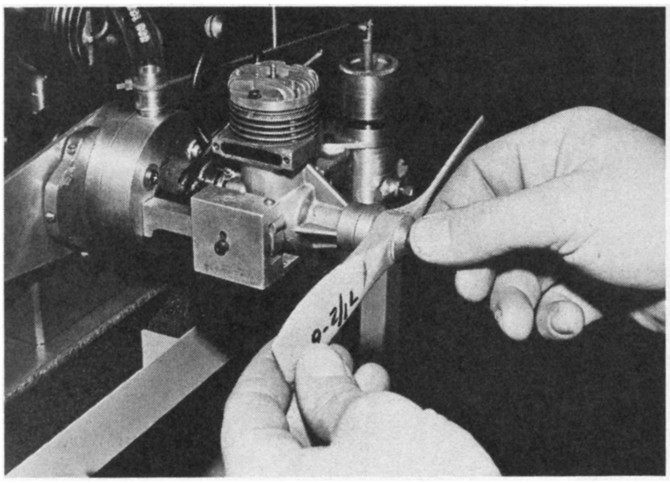
As we mentioned before, Scavenge efficiency is directly responsible for B.H.P. output. Also, scavenge efficiency is dependent upon the interior design of the engine layout. Let's take a brief look at some of these controlling factors and how they tend to affect our scavenge efficiency curve. This curve is based upon the scavenge efficiency, expressed as a per cent (%) and plotted against engine R.P.M. (see engine test data graph).

Before we begin I would like to point out that the *trial and error* method of modifying the inner workings of the engine offers a basis of comparison in scavenge efficiency tests. For example, the engine is tested in its stock condition, including R.P.M., torque and air-fuel flow rate. We now have our basis for comparison which is absolutely necessary. From this point on, the scavenge efficiency curve will be an indicator of design improvement.

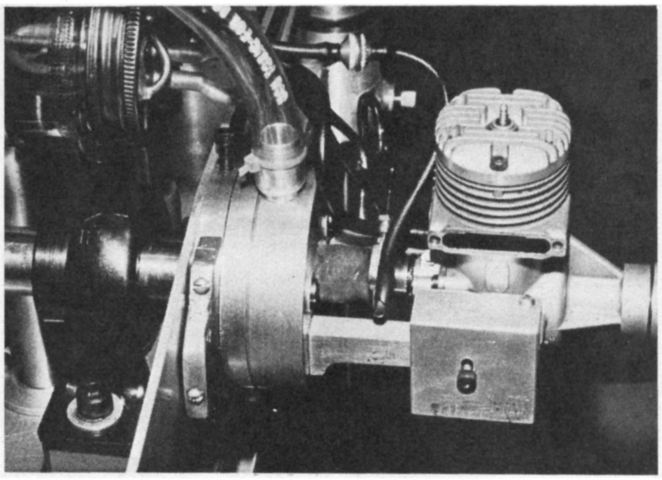
1. Type of scavenging system:
   Cross, Loop, Schnurlle,
   Schnurlle with Boost, etc.
2. Exhaust and Transfer Port Timing.



Long radius Flow Nozzles. The .40 engines utilize either the ⅜" or ½" I.D. nozzles. A larger ¾" nozzle is for .60 engines, while smaller ¼" nozzle's for proportionally smaller engines.



Placing a load on the test engine. The load of course is the propeller. Carefully record which prop etc. in your tests. Prop performance varies.



Pictured here is the air pumping into the carburetor. Transparent hose from torque arm block to engine carburetor. The arrangement worked well.

3. Crankcase compression ratio.
4. Induction valve timing.
5. Inlet and exhaust tuning.
6. Ducting efficiency.

The above areas represent the general breakdown of scavenge efficiency considerations within our engine. It is not the purpose of this series of articles to suggest specific internal engine modifications. Our method of analysis is of prime concern here.

As was seen, scavenge efficiency is affected by many variables. A very important result of good scavenge efficiency is discussed below.

*The density of the fresh charge after arrival in the cylinder.* As the fresh charge arrives in the hot cylinder, heat is transferred to it from the hot chamber walls and the hot residual exhaust gases, raising its temperature. This results in a decrease in the mass of fresh charge admitted and a reduction in scavenge efficiency. The scavenge efficiency is increased by low temperatures and high pressures in the fresh charge, since density is thereby increased, and a greater weight of charge can be inducted into a given volume. Thus, the power of an engine can be increased if its scavenge efficiency is improved by taking air from a point where it is *cooler* and/or by improving the induction timing, crankcase compression, transfer port timing, ram induction, etc., so as to *increase the pressure* at transfer port opening.

The accumulative effect of these factors are expressed on the scavenge efficiency curve as a *percent* of the maximum amount of air which could have been forced into the engine cylinder.

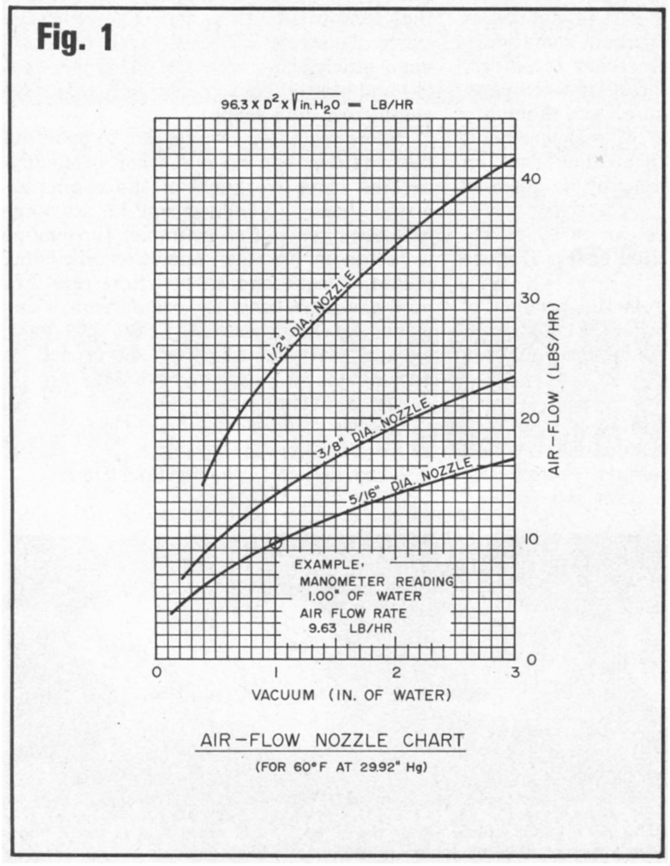
## B. Air-Fuel Ratio

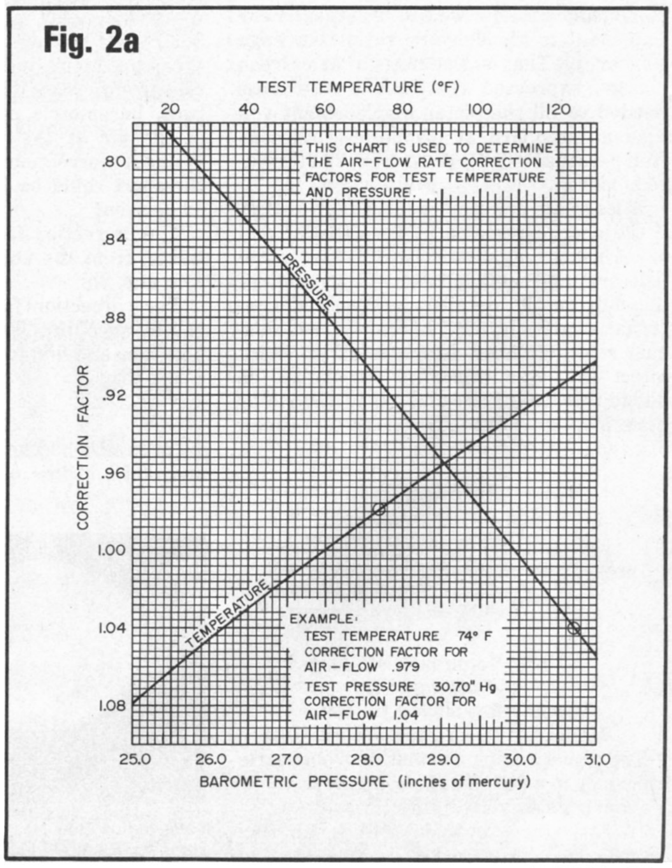
Air-Fuel ratio expresses the proportion of fuel and air, by weight, in a mixture.

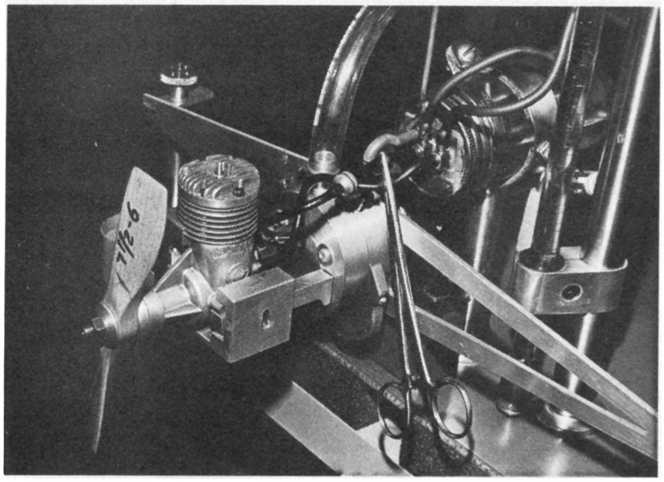
$$\text{Air-Fuel Ratio} = A/F = \frac{(\text{Lb. air/time})}{(\text{Lb. fuel/time})}$$

There is a definite limited range of air-fuel mixtures which are combustible. The proper proportions of fuel and air depend upon the *chemical composition,* of the fuel and the conditions under which the engine is operating.

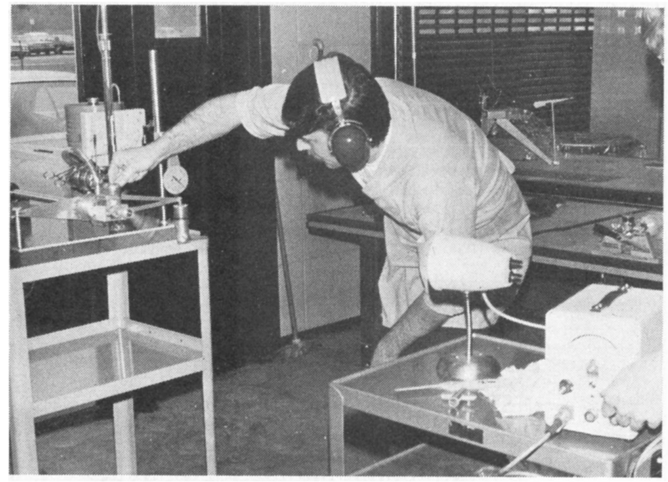
A mixture that contains just enough *air* to support complete combustion of all of





With the tank filled up, the fuel and pressure lines are pinched off to prevent "flooding" of the engine before the test. A methodical approach.



Here the needle valve is being adjusted to peak power before the data is collected. Note the ear protection, something for active modelers too.

the *fuel* is called a *chemically correct,* or *stoichiometric,* air fuel ratio. One that contains an excess of fuel is termed a "rich" mixture and one that contains an excess of air is termed a "lean" mixture.

Although it is possible to predict the amount of air necessary to obtain a stoichiometric combustion process within our engine by the chemical analysis of a hydrocarbon fuel, I believe you will agree that it is a rather impractical procedure! In actuality, our air-fuel mixture is usually set slightly "rich" at the full power needle setting at the carburetor. We soon learn through painful experience with ruined engines, not to squeeze the last R.P.M. from the needle setting. More often than not, the peak power setting will result in a "hot" run with possible physical damage resulting.

Other reasons for a richer than chemically correct mixture include:

a) Prevent overheating of critical engine parts by reducing flame and cylinder temperatures.

b) Reduces the possibility of detonation by reducing the flame temperature.

c) In the combustion of hydro-carbon fuels, it has been determined that the highest *flame speeds* will be obtained with an A/F ratio somewhat *richer* than chemically correct. (High flame speeds=increased rate of pressure rise, which influences peak cylinder pressures).

$$\text{Air/Fuel Ratio} = \frac{\text{Corrected Air-Flow (Lb's/Hr.)}}{\text{Corrected Fuel-Flow (Lb's/Hr.)}}$$

As was discussed last month, the *fuel-flow* chart is based upon the *specific gravity* of one (1), and a test consumption of a *10 m/volume.*

(See Figure #3).

## C. Other Calculations

From the air-fuel data, we could also calculate a number of other factors in order to develop a complete understanding of the total operating characteristics of our engine.

However, these other factors are of little help to us in improving engine power, or the more efficient application of this power to our models.

A brief explanation of these other factors will be made with the appropriate formula for those individuals interested in total engine analysis.

1. *Specific Fuel Consumption* (S.F.C.)

Specific fuel consumption is a measure of efficiency which indicates the amount of fuel an engine consumes for the work it produces.

$$\text{S.F.C.} = \frac{\text{Fuel Flow (Lbs/H.R.)}}{\text{B.H.P.}}$$

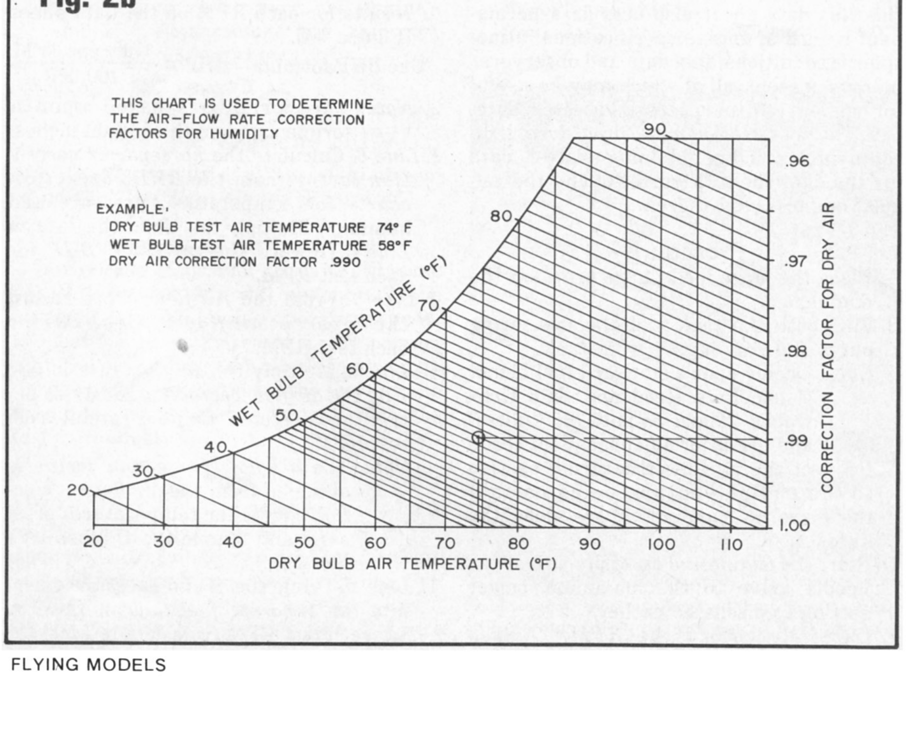
2. *Brake Thermo Efficiency* ($\eta$tb)

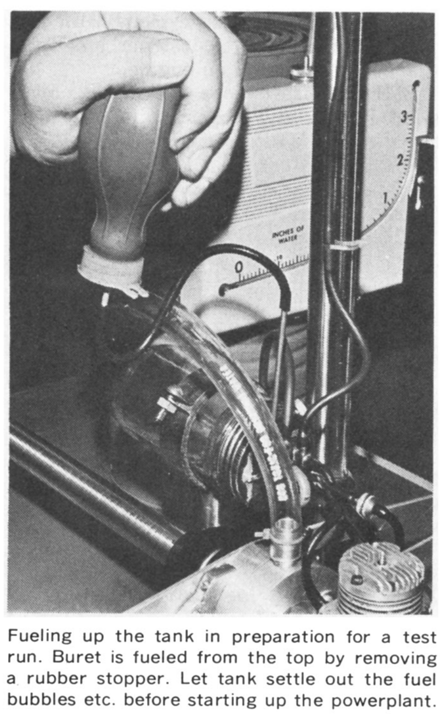
Brake Thermo efficiency is the ratio of energy available in the fuel to the energy equivalent of the B.H.P. delivered at the crankshaft.

$$\text{Brake Thermo Efficiency (\%)} = \frac{\text{Actual Output x 100}}{\text{Heat Input}}$$

Brake thermo efficiency takes into account all of the losses in the engine. For example, Thermo; mechanical.

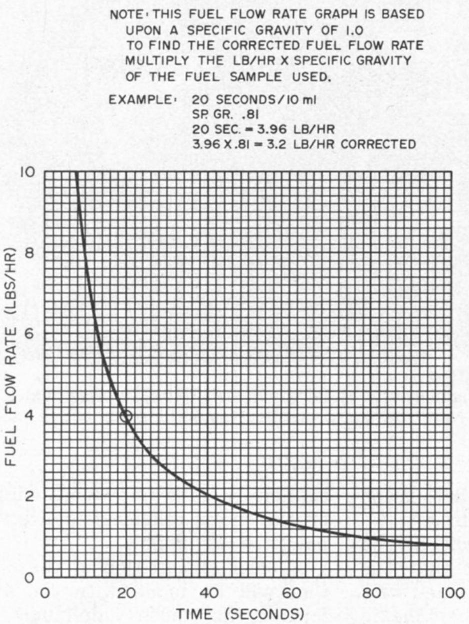
**Fig. 2b**

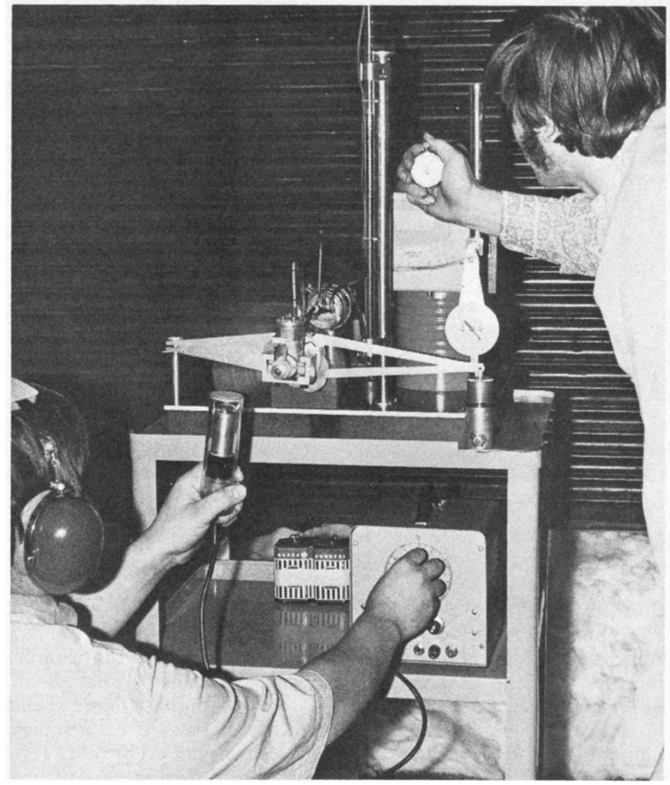


Fueling up the tank in preparation for a test run. Buret is fueled from the top by removing a rubber stopper. Let tank settle out the fuel bubbles etc. before starting up the powerplant.

Fig. 3



A neon stroboscope and stop watch are being used in collecting necessary data during the test. Notice the snow piled up under the overhead door.

The problem here involves finding the *heating value* of our fuel (heat input). Heating value can be found by actual tests in a calorimeter. Since calorimeter tests are costly, we have bypassed this factor. In order to express the *actual output* in the same unit, BTU, you use the conversion factor. This factor is: 1.41 BTU/Sec.=1 Hp. Brake Thermo efficiency is expressed as a per-centage (%).

*3. Brake-Mean-Effective-Pressure (BMEP)*

BMEP is the average effective pressure the engine can exert on the piston through one complete operating cycle. BMEP is a concept which helps a designer judge the safe power output of an engine.

BMEP is based upon the B.H.P., operating R.P.M. and physical dimensions of the engine. BMEP is expressed in terms of P.S.I. (pounds/square inch).

A simplified formula for calculating BMEP for our two-stroke cycle engines is:

$$\text{BMEP}=504{,}000\ \frac{\text{BHP}}{D^2\,\text{LMN}}$$

D=Diameter of piston (Bore)
L=Stroke
M=Number of cylinders
N=Engine R.P.M.

If you have been following along for the past three months, it has probably become obvious that conducting a complete engine analysis test is, at best, a time-consuming proposition. It is most imperative that accurate measurements are taken with a great deal of care in setting the needle valve on the engine. To recall for a moment, there are four (4) measurements to be taken at each engine load:

1. R.P.M.—tachometer
2. Ounces—torque scale reading
3. Fuel Consumption—stop watch (time required to consume 10 ml. of fuel)
4. Air consumption—Manometer reading

Again, at this point, I would like to re-emphasize the importance of an accurate tachometer. Small errors are magnified to a great extent in most of our calculations. Recommended are the Stroboscope or direct drive mechanical tachometers. The light-reflective type tachometers which are now on the market, have been found to be inadequate for our purposes.

## D. A Dynometer Test

1. In Figure #4 you will find a sample *engine analysis data sheet*. As the four measurements are taken from the test, they are entered onto this sheet. Notice this this data sheet also acts as a permanent record of engine specifications; atmospheric conditions; fuel data and observers/helpers present, all of which may be useful for comparison purposes at a later date.

2. The following *sample procedure* indicates the method of obtaining analysis data for the data sheet (Figure #4) and the calculations involved therein.

### Procedure

A. Place the large load (large prop) on the engine.

B. Fuel both the tank and the measuring buret and run engine from tank.
*Note:* Decide upon a standard fuel blend. (I use Pete Reed and Sam Griswald's "Tigre Sweat" (65% nitro) and use it for *all* subsequent tests.

C. Connect the air-flow measuring system to the engine venturi, being certain there are no leaks anywhere along the line.

D. Start the engine and carefully adjust the needle valve to the maximum power setting as discussed earlier.

E. Carefully record the *RPM; Ounces* (scale); *Manometer Reading* (air); and *Time* to consume 10 ml of fuel (fuel flow rate).
*Note: Open* flow valve from measuring buret and then *close* the flow valve from the tank.

F. Shut engine down. Close buret valve and open tank valve. Refuel Buret.

G. Change to the *next smallest load* on the engine and repeat steps D-F.

After the data has been collected for *all* of the test loads and entered on lines 1-4 of the *engine analysis data sheet* (Figure #4), we are ready to calculate the factors one lines 5-13.

H. *Line 5.* Calculate the *BHP* and enter the results for each RPM on the data sheet (Figure #4).
Use BHP formula—$\text{BHP}=\dfrac{\text{OuncesxRPM}}{100{,}000}$
*Note:* This formula is based upon a torque arm length of 10.00 inches.

I. *Line 6.* Calculate the *horsepower correction factor* from the *BHP correction charts* for temperature, pressure and humidity. Multiply this times the data on *Line 5.* Record the *corrected BHP* for each test RPM.

J. *Line 7.* From the *Air-flow chart* record the *apparent air flow* in Lbs./Hr. for each test RPM.

K. *Line 8.* Multiply the *air flow rate* in *line 7* by the *air flow correction factor* to obtain the *corrected air flow* rate at each test RPM.
*Note:* The *air flow correction factor* is obtained from the *air flow correction charts* for temperature, pressure and humidity. (Figure #2a and 2b).

L. *Line 9.* From the *fuel-flow chart* calculate the *apparent fuel flow* in Lbs/Hr. for each test RPM from data in *Line 4.*

**Fig. 4**

TEST No. ______ ENGINE ANALYSIS DATA SHEET TEST DATE ______

| | |
|---|---|
| ENGINE MANUFACTURER | HEAD TYPE |
| ENGINE DISPLACEMENT | CLEARENCE VOLUME (DROPS–OIL) |
| BORE AND STROKE | PLUG DEPTH |
| STROKE / BORE RATIO | DECK CLEARENCE |
| PORTING | HEAD DEPTH |
| SCAVENGING SYSTEM | HEAD CLEARENCE |
| INDUCTION SYSTEM | SQUISH BAND WIDTH & ANGLE |
| EXHAUST OPENS | FUEL |
| EXHAUST CLOSES | STANDARD MIX |
| TRANSFER OPENS | |
| TRANSFER CLOSES | |
| BOOST OPENS | SPECIFIC GRAVITY |
| BOOST CLOSES | ATMOSPHERIC CONDITIONS |
| INDUCTION VALVE OPENS | TEMPERATURE ______ °F |
| INDUCTION VALVE CLOSES | PRESSURE ______ IN. OF MERC. |
| CARB. CHOKE DIA. | WET BULB TEMP ______ °F |
| PIPE | |

TEST HELPERS / OBSERVERS

| | LOADS | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|
| 1 | TEST R.P.M. | | | | | | | |
| 2 | OUNCES (SCALE) | | | | | | | |
| 3 | AIR METER (IN./$H_2O$) | | | | | | | |
| 4 | FUEL FLOW RATE (SECONDS/10ml) | | | | | | | |
| | | | | | | | | |
| 5 | BRAKE HORSEPOWER (B.H.P.) | | | | | | | |
| 6 | CORR. HORSEPOWER | | | | | | | |
| 7 | AIR FLOW (LBS/HR) | | | | | | | |
| 8 | CORR. AIR FLOW | | | | | | | |
| | | | | | | | | |
| 9 | FUEL FLOW (LBS/HR.) | | | | | | | |
| 10 | CORR. FUEL FLOW (SP. GR. X #9 | | | | | | | |
| 11 | TORQUE (OZ.–IN.) | | | | | | | |
| 12 | AIR / FUEL RATIO | | | | | | | |
| 13 | SCAVENGE EFFICIENCY (%) | | | | | | | |

TRUE TORQUE (F X r) IS FOUND BY MULTIPLYING THE FORCE IN OUNCES (#2) TIMES THE TORQUE ARM LENGTH (r) OF 10" FOR THIS DYNAMOMETER.

**Fig. 4**

TEST No. ______ ENGINE ANALYSIS DATA SHEET TEST DATE FEB. 14

| | |
|---|---|
| ENGINE MANUFACTURER BRAND X | HEAD TYPE — |
| ENGINE DISPLACEMENT .40 | CLEARENCE VOLUME (DROPS–OIL) — |
| BORE AND STROKE — | PLUG DEPTH — |
| STROKE / BORE RATIO — | DECK CLEARENCE — |
| PORTING — | HEAD DEPTH — |
| SCAVENGING SYSTEM — | HEAD CLEARENCE — |
| INDUCTION SYSTEM — | SQUISH BAND WIDTH & ANGLE — |
| EXHAUST OPENS — | FUEL |
| EXHAUST CLOSES — | STANDARD MIX 65% NITRO |
| TRANSFER OPENS — | METHANE |
| TRANSFER CLOSES — | |
| BOOST OPENS — | SPECIFIC GRAVITY 1.05 |
| BOOST CLOSES — | ATMOSPHERIC CONDITIONS |
| INDUCTION VALVE OPENS — | TEMPERATURE 70° °F |
| INDUCTION VALVE CLOSES — | PRESSURE 28.95 IN. OF MERC. |
| CARB. CHOKE DIA. — | WET BULB TEMP 60° °F |
| PIPE — | * 3/8" AIR FLOW NOZZLE |

TEST HELPERS / OBSERVERS

| | LOADS | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|
| 1 | TEST R.P.M. | 22,000 | 20,800 | 19,000 | 18,000 | 17,000 | 16,300 | |
| 2 | OUNCES (SCALE) | 8.6 | 9.1 | 9.7 | 9.8 | 9.6 | 8.9 | |
| 3 | AIR METER (IN./$H_2O$) | .73 | .87 | .90 | .86 | .79 | .63 | |
| 4 | FUEL FLOW RATE (SECONDS/10ml) | 18.5 | 17.9 | 17.8 | 17.9 | 18.4 | 19.8 | |
| | | | | | | | | |
| 5 | BRAKE HORSEPOWER (B.H.P.) | 1.89 | 1.90 | 1.84 | 1.76 | 1.63 | 1.45 | |
| 6 | CORR. HORSEPOWER | 1.98 | 2.00 | 1.93 | 1.85 | 1.71 | 1.52 | |
| 7 | AIR FLOW (LBS/HR) | 11.66 | 12.73 | 13.00 | 12.71 | 11.99 | 10.93 | |
| 8 | CORR. AIR FLOW | 10.50 | 11.46 | 11.70 | 11.44 | 10.79 | 9.84 | |
| | | | | | | | | |
| 9 | FUEL FLOW (LBS/HR.) | 4.25 | 4.38 | 4.44 | 4.39 | 4.29 | 4.02 | |
| 10 | CORR. FUEL FLOW (SP. GR. X #9 | 4.46 | 4.60 | 4.66 | 4.61 | 4.50 | 4.22 | |
| 11 | TORQUE (OZ.–IN.) | 86 | 91 | 97 | 98 | 96 | 89 | |
| 12 | AIR / FUEL RATIO | 2.35 | 2.49 | 2.51 | 2.48 | 2.40 | 2.33 | |
| 13 | SCAVENGE EFFICIENCY (%) | 45 | 52 | 58.1 | 60 | 59.9 | 57 | |

TRUE TORQUE (F X r) IS FOUND BY MULTIPLYING THE FORCE IN OUNCES (#2) TIMES THE TORQUE ARM LENGTH (r) OF 10" FOR THIS DYNAMOMETER.

M. *Line 10.* Multiply the *specific gravity* of the test fuel times the *apparent fuel flow* on *Line 9* to obtain the *corrected fuel flow* for each test RPM.

N. *Line 11.* To obtain *true torque,* multiply the factors from *line 2* times the *torque arm length* of the dynamometer for each test RPM as expressed in Oz.-Inches.

O. *Line 12.* Calculate the *Air/Fuel ratio* by dividing the *corrected air flow* from *line 8* by the *corrected fuel flow* from *line 10* for each test RPM.

$$\text{Air/Fuel Ratio} = \frac{\text{Corrected Air Flow}}{\text{Corrected Fuel Flow}}$$

P. *Line 13.* Calculate the *scavenge efficiency* by utilizing the factors from *line 8 (corrected air flow),* from the engine's *cubic inch displacement* and each test RPM in the manner shown :

$$\text{Scavenge Efficiency (\%)} = \frac{37{,}750 \text{ x corrected Air Flow}}{\text{Engine in}^3 \text{ x RPM}}$$

3. Plot the results of all test data obtained on the *Test Data Graph* (Figure #5).

## E. Analysis of Air Fuel Flow

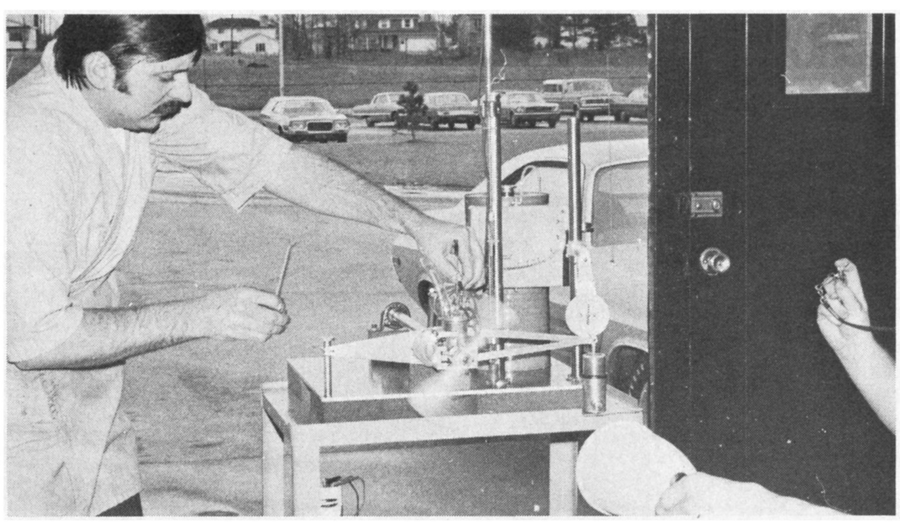
In the case of air/fuel ratios (A/F) some very interesting data will undoubtedly be obtained for each engine operating speed. It must be remembered that the correct proportions of fuel and air depends upon the chemical composition of the fuel and the conditions (atmospheric) under which the test is conducted.

### Fuel

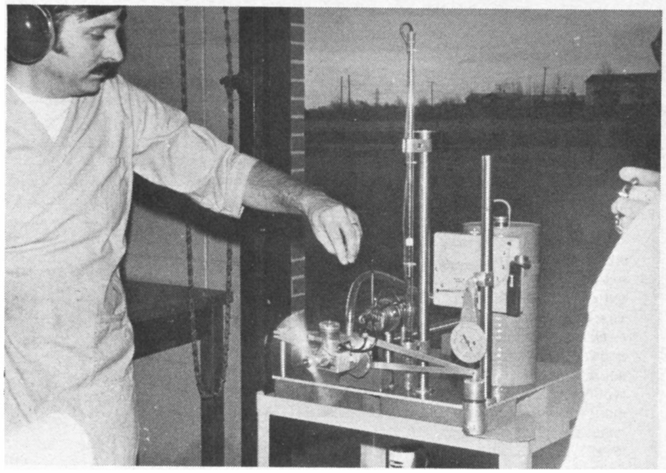
What's in the can? If you are at the back of the pack, nobody cares. But, if you're among the first to get the checker, the contents of your tank become very interesting to all.

At first glance, Nitro Methane appears to be a poor choice for our fuels. From the aspect of thermo output, observe column A below, where Nitro Methane is compared with other common fuel ingredients:

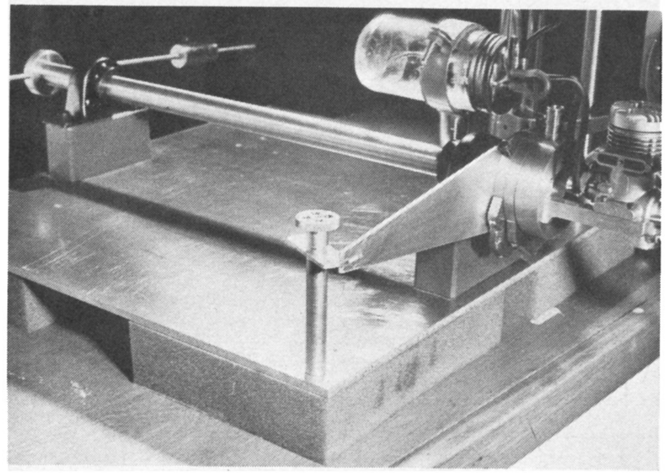
| Fuel | BTU/LB. A | x F/A = | BTU/LB. Air Inducted B | Stoichio-Metric A/F | Peak Power A/F |
|---|---|---|---|---|---|
| Gasoline | 20,000 | .066 | 1320 | 15:1 | 12.5:1 |
| Methanol | 9,600 | .154 | 1386 | 6.5:1 | 4.5:1 |
| Nitro Methane | 5,000 | .588 | 2940 | 1.7:1 | — — |
| Nitro Benzene | 10,800 | .143 | 1544 | 7:1 | — — |
| 1-Nitro Propane | 10,000 | .172 | 1720 | 5.8:1 | — — |
| Nitro Ethane | 8,000 | .238 | 1904 | 4.2:1 | — — |
| Kerosene | 19,000 | .066 | 1254 | 15:1 | — — |
| Ethanol | 11,500 | .111 | 1277 | 9:1 | 6.5:1 |
| Propylene Oxide | — — | .105 | — — | 9.5:1 | — — |
| Benzene | 17,000 | .076 | 1292 | 13.2:1 | 11:1 |



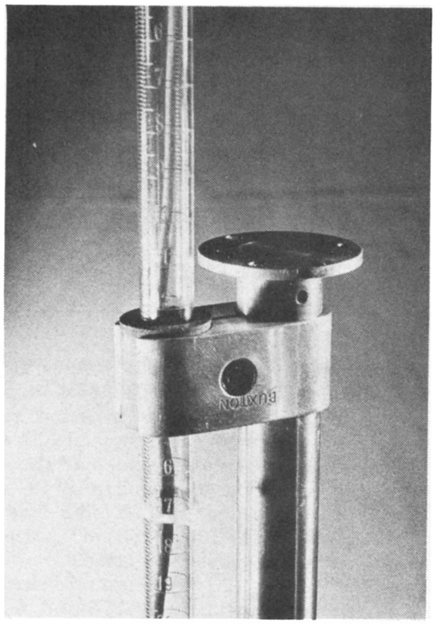
Fuel system is being switched from tank to buret for consumption measurement. Note strobe being utilized along with a stop watch. Part of Dave's experimental program at West Seneca Jr. High.



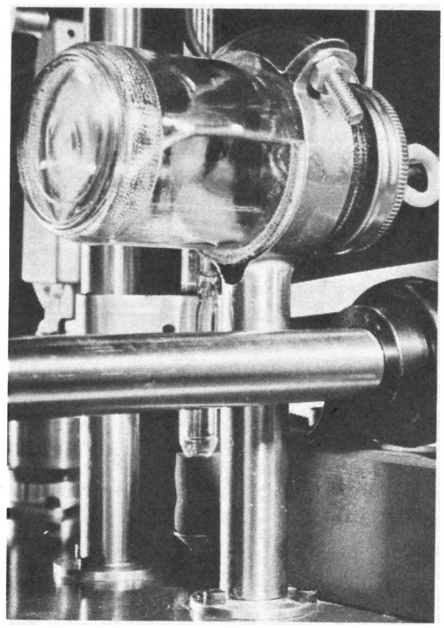
Torque arm "for stop" has just been released in preparation for a torque reading to be taken. Results showed consistent, meaningful, useful data.



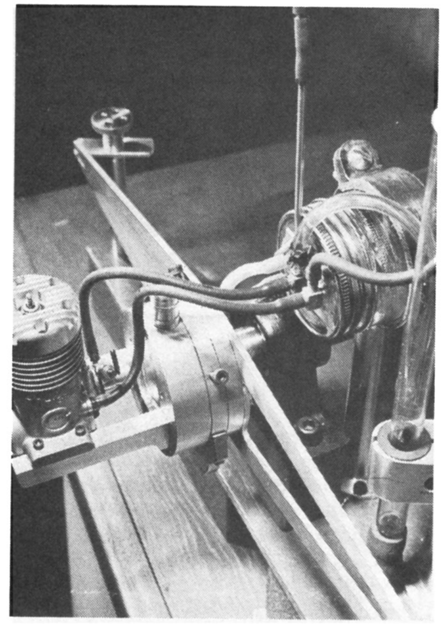
The little construction details. Torque arm "stop" settings and the main torque/shaft bearings. All must work smoothly and in unison to perform.



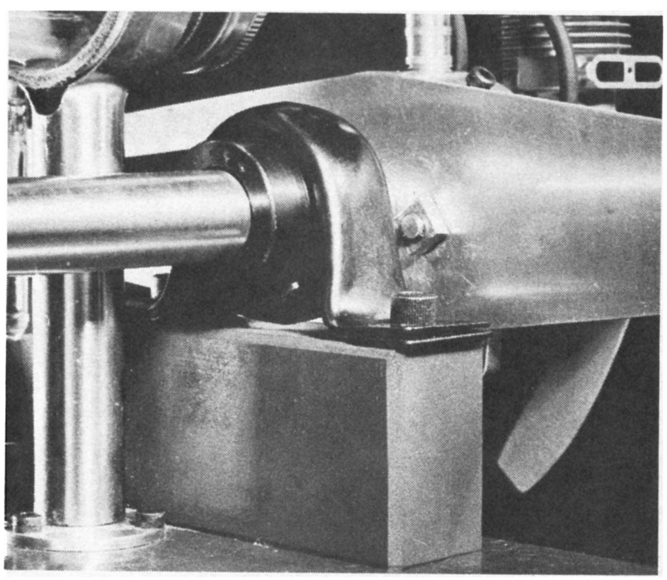
Buret clamp device. Notice the rubber tubing is around the Buret for vibration absorption. This helps give more accurate and meaningful result in final analysis. Give your all for accuracy.



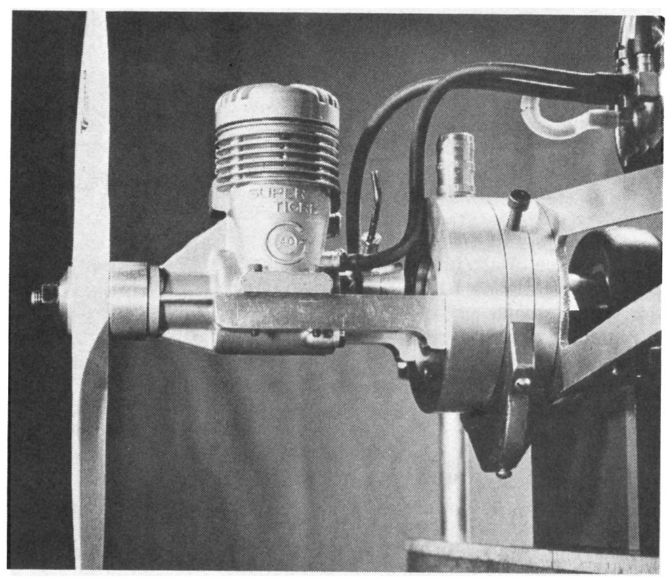
Tank-glass jar (8 oz.) elevated and supported by stainless steel tubing, support post. Notice the leather friction grip under the tank strap. A gasket must be supplied under the jar's lid.



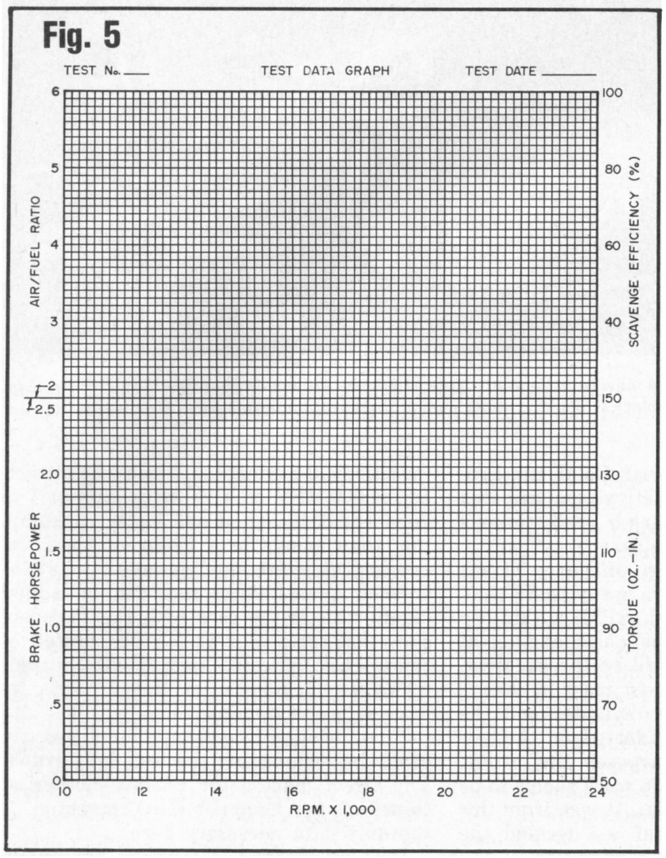
Dynamometer with torque arm and balancing block in evidence. Notice the fuel and pressure lines entering the blocks (junction) on the tank lid. All plumbing connections should be tested well.

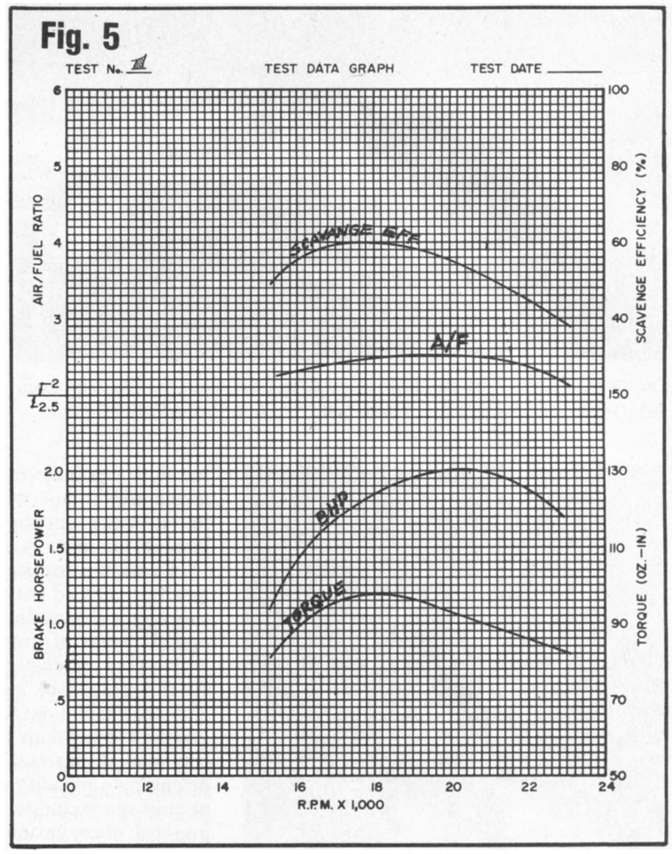


A close-in look at the torque arm, the bearing and the pillow block. The students in this pilot school program became completely engrossed in it.



A good view of the torque arm block, with motor mounts attached. Notice static balance weights attached to torque arm. See fuel-pressure lines.




From these figures it can be seen that Nitro Methane simply doesn't fill the bill as a high thermo output fuel. Actually, it compares closely to the heat energy liberated by the burning of wood!

Not understanding why nitro methane would ever be considered as a fuel, we initiated the first air-fuel measurements on a test engine. To my disbelief, the air-fuel ratios obtained for the 65% nitro methane test fuel were only about 2.5:1! This I guessed could not be possible since the chemically correct A/F ratio for gasoline was close to 15:1. How could we be running 2.5:1? Subsequent tests revealed these ratios to be correct. The air flow measurements were suspected of being fouled up. Upon checking the air flow data in the scavenge efficiency formula, it was found not to be at fault, with efficiency running about 60%.

We found that if we ran a test on alcohol/castor oil fuel the A/F ratio increased to about 5:1. The perplexing fuel problem remained just that for about six months while torque-horsepower analysis continued successfully. During this time, the perplexing situation was discussed with literally dozens of people without any satisfactory conclusions reached.

Finally, in one of those infrequent brainstorms, the solution became apparent and later, very obvious. Let's use Methanol and Nitro Methane as examples:

Remember that we said an engine's power potential is directly related to the amount of air which may be inducted in a given period of time. The greater the air flow, the greater the amount of fuel which may be added, stochiometrically. This results in a greater thermo energy consumption per unit time, which increases the resultant Brake Horsepower.

Alcohol has a stoichiometric A/F ratio of 6.5:1 while nitro methane is 1.7:1. In order to compare these two fuels from the aspect of amounts added to a unit weight of air, it is convenient to use the reciprocal of A/F, or the F/A ratio. The F/A ratio for nitro methane is .558:1. The F/A ratio for methanol is .154:1. What this means is that nitro methane can induct approximately 3½ times as much mass per unit time than methanol. In other words, nitro methane combines and burns chemically correct with about twice its weight of air. Alcohol on the other hand, admittedly of higher BTU value, only combines and burns chemically correct with about 6 times its weight of air. Compare the various fuels from the aspect of BTU/Lb. of air inducted as shown in column B.

By studying the chart, you can obtain an idea of the relative heat values of various fuels, in comparison to nitro-methane. Nitro methane certainly shows up more favorably when viewed from the aspect of inducted thermo energy/period of time.

In general, the best fuels contain the highest proportion of Hydrogen, since it has 62,000 BTU/Lb. while carbon has only 14,500 BTU/Lb.

In data obtained from "Liquid Propellant Rockets," the theoretical decomposition products in Mole fractions for nitro methane was obtained. Acting as a Monopropellant, these are:

$CO_2$ — .057
CO — .277
$H_2O$ — .277
$H_2$ — .223
$N_2$ — .116

When used as a monopropellant, nitro methane does not have a sufficient amount of oxygen to oxidize the CO and $H_2$ that is formed.

Because we have an excess of air, it is believed that in a piston engine the CO and $H_2$ would be oxidized to $CO_2$ and $H_2O$. Therefore, the products of combustion for $CH_3NO_2$ would probably be $CO_2$, $H_2O$, $N_2$

$$4\,CH_3NO_2 + 3O_2 \rightarrow 4\,CO_2 + 6H_2O + 2N_2$$

Oxygen from the nitro methane molecules is liberated and used to oxidize the Carbon and Hydrogen.

It is now possible to find the BTU input of these fuels in order to calculate the engine's total or brake thermo efficiency ($\eta tb$). In the case of Methanol/caster oil fuels, the weight of the oil may be subtracted from the total, leaving the actual weight of the Methanol (or actual thermo potential) to be measured flowing into the engine (Lb/Hr).

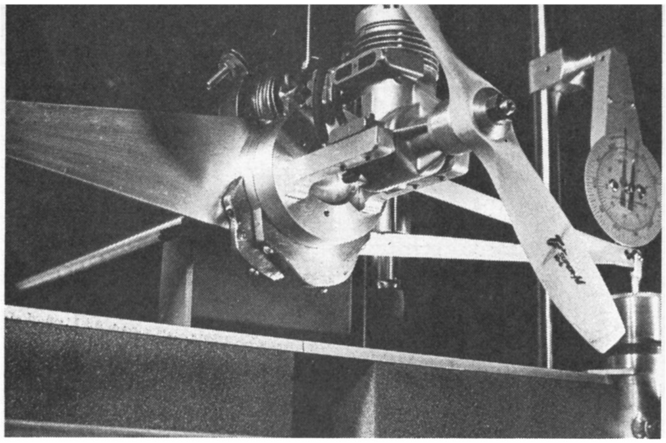
*Example:* 3.4 Lb.—Actual Methanol/Hr. at 15,000 RPM
9,600 BUT/Lb.—Methanol
3.4 Lb./Hr.=32,640 BTU/Hr. (Input)
BHP=1.0 at 15,000 RPM
1 Hp.=1.41 BTU/Sec. (As discussed before)
1 Hp.=1.41x60x60=5076.0 BTU/Hr. *Output*

$$\eta tb = \frac{\text{Output}}{\text{Input}} \times 100$$

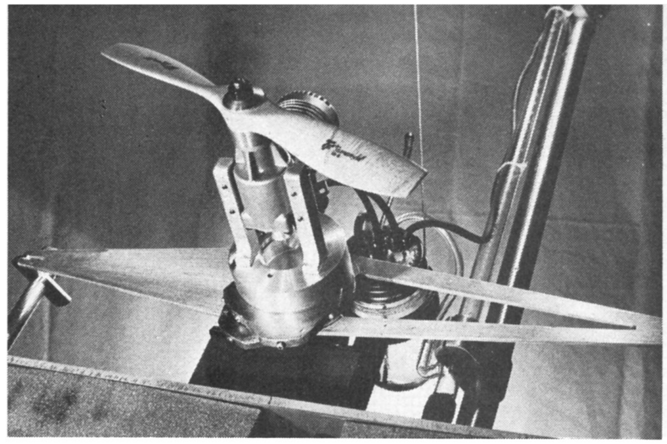
$$\eta tb = \frac{5076.0\ \text{BTU/Hr.}}{32{,}640\ \text{BTU/Hr.}} \times 100$$

$$\eta tb = 15.5\%\ \text{at}\ 15{,}000\ \text{RPM}$$

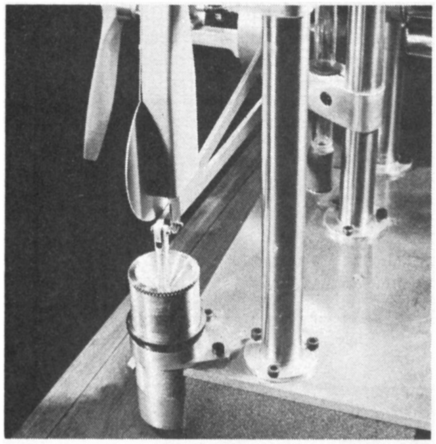
Special thanks are extended to Ralph Furness for his invaluable assistance deal-



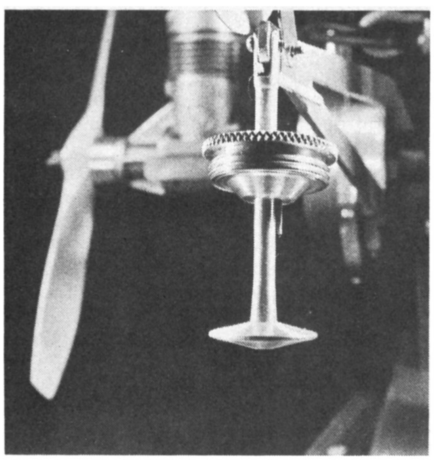
Excellent view showing the balance weights, base plate, torque arm, foam absorption unit, scale, etc. Strive for accuracy, seek out the reasons.



A view from beneath: Notice here the expansion chamber in the torque arm block which does not have its carburetor adapter plate attached to it.



Hydraulic dampener assembly does not affect force readings when equilibrium is reached. Below is magnesium piston which moves through 10-viscosity oil to resist rapid directional change.



ing with the fuels material. Ralph is a Professional Chemist who actively participates in our great hobby. Ralph really enjoys mixing business with pleasure!

In relation to scavenge efficiency, it has been observed through many tests that there is a certain speed (RPM) within the speed range of any particular engine, at which this *air* charge will be the greatest. At this point, the greatest mass of charge can be packed into the cylinder and the greatest force can be exerted on the piston. For all practical purposes, the torque, or engine's capacity to do work seems to be at this approximate point. It was from this general observation that we decided the *maximum acceleration* capabilities of the given engine would be located at the *maximum torque point* on the RPM range.

As the speed of an engine is increased above the speed for maximum scavenge efficiency (maximum charge/cycle) the quantity of charge will drop off (similar to torque). However, since the engine speed is increasing, more charges will be inducted during a given period of time. The air consumption or quantity of air swallowed during a given period of time continues to increase, as indicated by the Manometer readings with decreasing engine loads.

This bit of data confused me because if more air can be inducted per period of time, so should more fuel be added. After more research on the matter, it was discovered to be a correct observation and the answer presented itself: Air consumption will continue to increase with increased engine speed until some point is reached where the charge per stroke is dropping off more rapidly than the number of strokes per time is increasing. This dropping off point is somewhere beyond the operating speed (practical) of any engine which I have tested (24,000 RPM on a .40 is as high as I've gone and "chickened out.") Air consumption then brings in a factor of time. Increased air consumption means that increased quantities of fuel can be added (again, this is the case with the fuel consumption tests) during a given period of time. This indicates that power output, which also includes a time factor, can be increased proportionately.

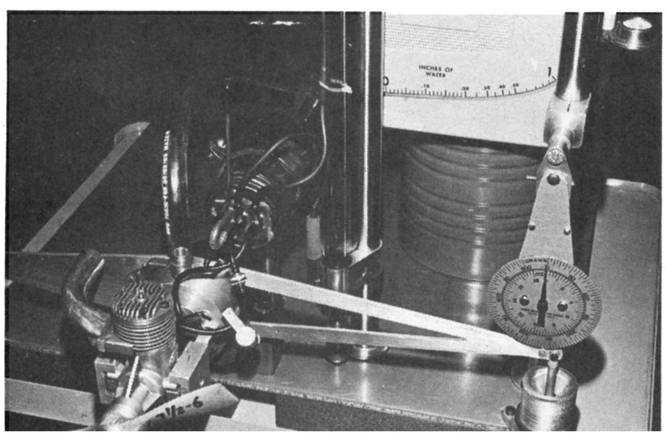
If you wish, air consumption curves can easily be included on the test data graph. The direct manometer reading converted to pounds per hour (Lb./Hr.) corrected, is the only data necessary here.

This information comparing the maximum charge per cycle (peak scavenge efficiency) and the peak air consumption point with its time factor seems to be an approximate comparison between torque and horsepower. Actually, the air consumption figures more closely relate to the indicated horsepower curve which does not consider mechanical (frictional) losses. The drop in air consumption may, therefore, be attributed to the increasing frictional losses at higher speeds. BHP=IHP—FHP at any given RPM.

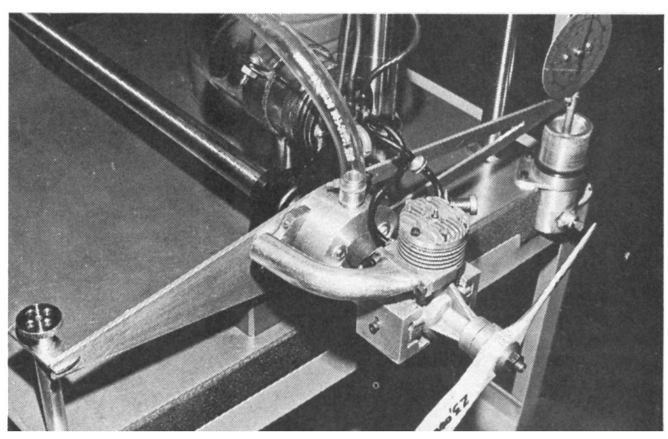
F.) After a period of time during which you become familiar with basic engine tests with their comparisons, many specific tests may be undertaken. Some of these tests include the following:

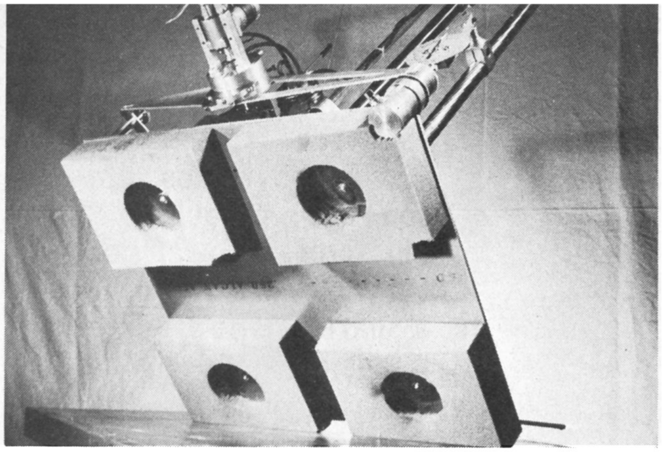
- a) Fuel Tests
  1. Peak BHP
  2. Scavenge Efficiency
  3. A/F ratios for peak power point
- b) Glow Plug Tests
- c) Compression ratio Tests
- d) Intake & Exhaust Tuning
  1. Effect on BHP Peak

An engine with a Mini-Pipe attached for test purposes. Front-left view. Results go into the spin-off of data which accumulates in the notebook.

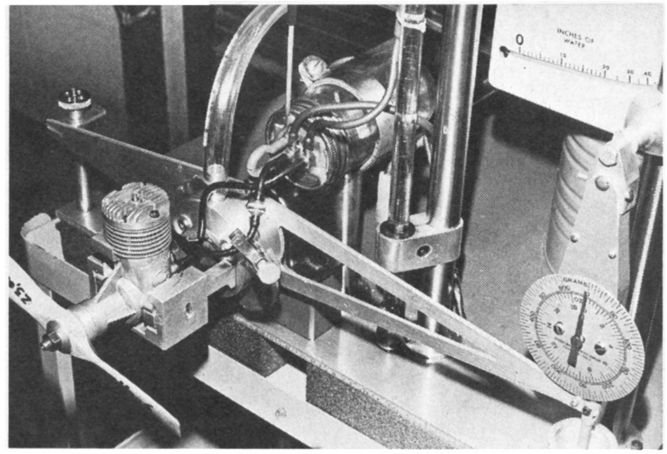


As seen from front-right. Before each trial, a check to make sure all is tight and secure. Any such variable would alter test result quite a bit.





Another bottom view of the foam vibration dampening device. Notice a hydraulic dampener. The engine must run smoothly for these test results.



General assembly view of the dyno. Dave poured many months of effort on the school program and project. It sheds some new light on old problems.

2. Effect on scavenge efficiency
3. Effect on Torque
4. A/F Ratio—Peak Power Point

## G. Instrumentation Construction Notes

1. *Dynamometer*

Most of the construction details should be self-explanatory after observing the included photographs. There are a few particulars which require an explanation, which will be discussed at this time.

a) *Hydraulic Vibration Dampener.* This piston and cylinder arrangement operates in an environment of low viscosity oil (10) which effectively dampens most of the vibrations from the engine to the spring scale, without absorbing any mechanical energy. The piston is fit loosely to the cylinder allowing the oil to slowly pass from one side of the piston to the other. Piston fit—.080 approximate clearance.

b) *Adjustable Lead Torque Arm Weights.* These weights allow the movable assembly to be static balanced about the torque arm pivot point. This effectively eliminates any outside forces from acting upon the sensitive torque arm scale. The assembly may be rebalanced for different engines or the addition of hardware from the engine (tuned pipe).

c) *Torque Arm Lock Post.* This locking device is found on the opposite end of the torque arm from the spring scale. When starting the engine, the torque arm is locked securely for obvious reasons. After the needle valve has been adjusted, the lock is easily released.

d) *Universal Engine Mount.* Some sort of adjustable mount should be devised in order to offer flexibility for testing a variety of engines without major design changes in the test unit. (See Photo).

e) *Foam Base for Dynamometer.* A very effective vibration absorption method for this unit is sheet foam contact cemented to the aluminum base. (See Photo).

2. *Air Flow Unit*

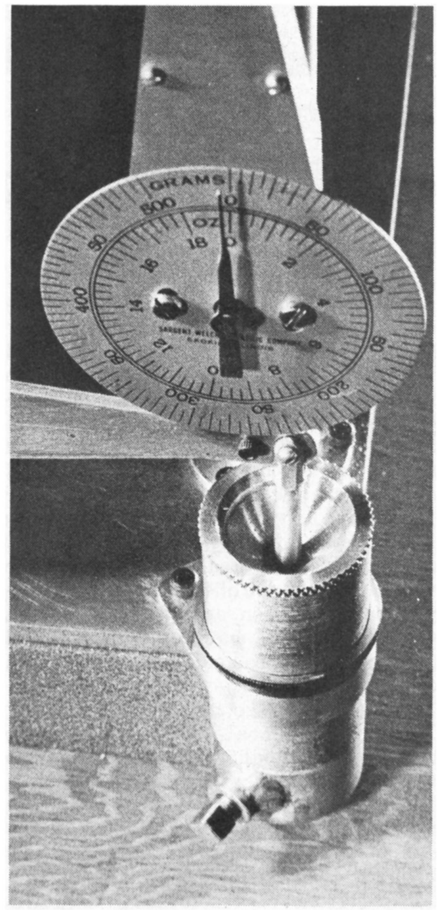
a) *Manometer.* This unit was purchased and may be obtained in most areas where industrial instrumentation is available.

b) *Dampening Drum (Pulse).* This unit was constructed from two coffee cans soldered together (end to end). The size of this unit is not important as long as it is not too small. The air to carburetor fitting was installed (soldered) as well as the long radius flow nozzle fitting. A tube from the drum was also installed for the Manometer. The photographs should be sufficient in answering your questions.

c) *Long Radius Flow Nozzles.* These air induction fittings were machined to specific diameters corresponding to various air-flow rates likely to be encountered in analysis work. Refer to the air-flow nozzle graph for an idea of the air flow ranges encountered. The machined radius of the nozzle is not important providing that a smooth flow passage is provided, thus reducing turbulence and resistance to flow. Care must be taken to ensure an airtight system be maintained.

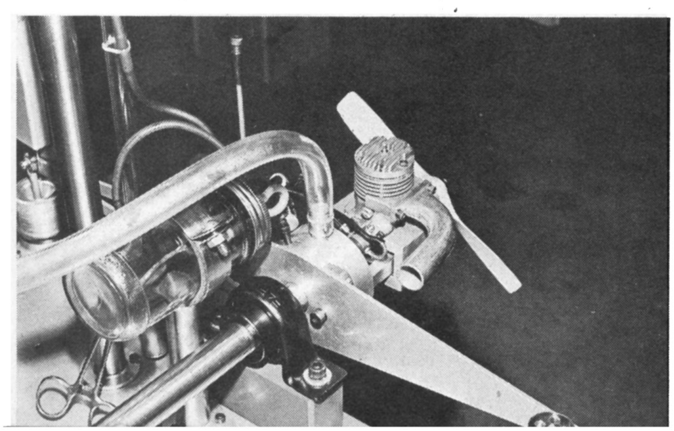
It is my hope that this series has offered some insight into the complicated area of engine analysis. The research, testing and evaluation necessary for this has taken better than a year in preparation and writing.

I might say, in closing that if it were not for inovator-machinist, Al Mahaffey, this project may never have been accomplished. Many thanks are extended to this very patient modeler-friend who tirelessly extended himself toward success of this venture.




The view of the torque arm scale and hydraulic dampener assembly, intact. When in doubt, take a second reading. Scales should be as accurate as possible for this type of experimentation.

A rear right view might put it all into perspective for you. Needless to say the hose fittings should all be free of leaks, devoid of kinks etc.



Dyno Stroboscope. The pipe adds .10 to .25 BHP, with proper design. Laboratory set-ups seek to find the answers that escape us in the field.